CAE



# Ammonia Application to Reciprocating Engines

FINAL REPORT

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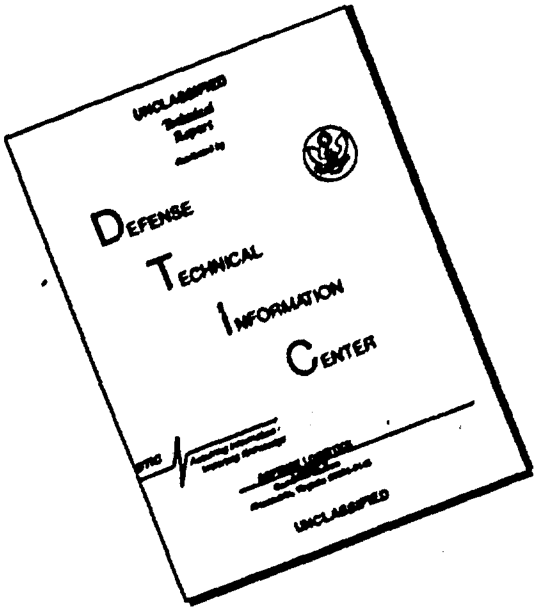


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CONTINENTAL AVIATION AND ENGINEERING CORPORATION
DETROIT, MICHIGAN

# AMMONIA APPLICATION TO RECIPROCATING ENGINES

FINAL TECHNICAL REPORT
CONTRACT DA-20-113-AMC-05553(T)

This work was performed under the technical supervision of the Detroit Arsenal, Warren, Michigan, under Contract DA-20-113-AMC-05553(T), administered by the Detroit Procurement District, 1580 East Grand Boulevard, Detroit, Michigan, 48211.



Reported By

T. J. Pearsall
Thomas J. Pearsall

Approved By

Ross L. Fryer
Ross L. Fryer

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## SUMMARY


Two engines, one naturally aspirated spark-ignition and one supercharged compression-ignition, were successfully converted to operate with anhydrous ammonia as fuel in lieu of the hydrocarbon fuel normally used.

Conversion of the spark-ignition engine required the following changes:

1. Increase of compression ratio.

2. Provide an ammonia carburetor.

3. Provide a high energy ignition source (magneto).

4. Provide long reach spark plugs.

The final configuration of the spark-ignition engine was able to deliver a maximum of 41 horse-power on ammonia alone and 53 horsepower on ammonia plus 1.5 percent hydrogen. The maximum output with gasoline was 65 horsepower. Higher outputs on ammonia were available but performance had to be curtailed because of high firing pressures and the danger of failing the cast iron crankshaft in the particular engine tested.

The compression-ignition engine was converted to operate with ammonia fuel in two different configurations. The first version was achieved by simply supplying ammonia vapor in the induction air system and using the fuel injection system to furnish a pilot charge for ignition purposes. With this arrangement the engine was able to produce as much as 132 percent of diesel fuel output while operating within the same limits of exhaust gas temperature and peak cylinder pressures. The quantities of diesel pilot fuel were as low as two cubic millimeters per stroke.

A second version of the compression-ignition engine to run on ammonia fuel was a conversion to spark-ignition by doing the following:

1. Remove the fuel injection pump and install a magneto.

2. Remove the fuel injection nozzles and install long reach spark plugs.

3. Provide for aspiration of ammonia vapor into the induction air system.

## SUMMARY

Performance with the spark-ignition conversion exceeded that with diesel pilot fuel. Effect of varying the compression ratio was also investigated.

Direct injection of liquid ammonia into a high compression ratio (30:1) engine was attempted and was unsuccessful.

Various other auxiliary aids such as fuel additives, ionization and radio frequency dissociation were also investigated.

A total of 1128 engine test hours were accumulated on the spark-ignition engine of which 1045 were with ammonia fuel; 954 hours (911 with ammonia) were accumulated on the compression-ignition engine.

## FOREWORD

Logistics studies of Army operations in World War II and Korea established that approximately 65 percent of the total tonnage required for support of combat operations consisted of fuels and lubricants. To compound this already heavy logistical burden, future Army concepts envision increased mechanication and greater emphasis on mobility and dispersion. Faced with these problems, the Army searched for other materials and devices for vehicle propulsion. Nuclear energy seemed to be the apparent answer.

Analysis proved that direct use of nuclear energy presented serious problems in application to vehicles. Attention was then turned to other potential applications of a nuclear energy power source. The Army, in a cooperative research and development effort, established the practicality of mobile nuclear reactors as a source of energy in the field. These studies indicated three possible approaches wherein nuclear energy could be used with direct or indirect energy conversion devices, or as the power source for the manufacture of chemical fuels. Further studies indicated that to realize early payoff the latter approach held the most promise. It was decided that a nuclear power source could provide the energy to synthesize chemical fuels with air and water as the on-site raw materials. This concept to provide on-site manufacturing of fuels is referred to as the Mobile Energy Depot, Reference 1. Materials showing the greatest potential were hydrogen, ammonia, hydrazine and hydrogen peroxide. Factors concerned with physical

# FOREWORD

and chemical properties, handling, storage and dispensing of the four fuels led to the choice of ammonia as the fuel with the greatest potential.

This contract provides for an investigation to determine the feasibility and practicality of anhydrous ammonia as a reciprocating engine fuel suitable for fulfilling Army operational requirements.

In accordance with the original terms of the contract, this program was scheduled to include the following items of work:

1. Investigate additives to ammonia fuel that can improve the ignition characteristics and flame propagation rate of ammonia in reciprocating engines.

2. Establish comparative engine performance of the L-141 engine on gasoline and ammonia fuels.

3. Design and develop a dissociator subsystem and test on the L-141 engine.

4. Determine effect of physical size and other design variables of the AOSI-895-5 engine on use of ammonia fuel in spark-ignition engines by adapting it to operate with ammonia.

5. Investigate the problems associated with use of ammonia in a compression-ignition engine, particularly in regard to ignition, combustion control, injection equipment, starting, combustion systems, and conversion to spark-ignition.

6. Investigate the effect of ionization on combustion in a compression-ignition engine.

7. Establish comparative engine performance of the LDS-465-1 engine on diesel and ammonia fuel.

8. Determine the effect of physical size and other design variables of the AVDS-1790-2A and the 8V71T engines on use of ammonia fuel in compression-ignition engines by adapting them to operate with ammonia.

## FOREWORD

9. Conduct endurance tests to determine the durability of each engine converted to operate on ammonia fuel.

A revision to the contract was issued by the Army Tank-Automotive Command to eliminate the durability tests and to reduce the number of engines to be adapted to operate on anhydrous ammonia. This was done to permit Continental to direct more attention to the details of a practical solution to burning ammonia in both spark-ignition and compression-ignition engines.

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## OBJECT

To evaluate the feasibility of using anhydrous ammonia as an alternate fuel for reciprocating engines in military equipment by:

1. Applying existing knowledge to burn ammonia fuel in a spark-ignition engine in a practical manner and improving the engine so that it will be equal in output and flexibility to an engine operating on gasoline fuel.

2. Developing concepts and demonstrating compression-ignition engine operation on ammonia fuel.

3. Demonstrating the effects of various bore sizes, combustion chamber shapes, and basic design on the ability to burn ammonia fuel.

## CONCLUSIONS

1. It is feasible and practical to use anhydrous ammonia as a fuel in reciprocating engines in remote temperate or tropic areas where the ammonia can be produced by a Mobile Energy Depot and where no hydrocarbon fuel is available. Operation in extremely cold climates may be very difficult.

2. Spark-ignition engines can be readily converted to operate with anydrous ammonia fuel in lieu of hydrocarbon fuel.

3. Increasing compression ratio, increasing ignition energy, and locating spark-plug electrodes near the center of the combustion chamber are the major changes necessary to convert a spark-ignition engine to operate with anhydrous ammonia as a fuel. Structual limits of existing engines must be considered because of the higher firing pressures.

4. Output of the unsupercharged spark-ignition, ammonia-fueled engine is limited to 80 percent of the gasoline-fueled engine output because of the heating value of the fuel.

## CONCLUSIONS

5. Due to the low rate of flame propagation, performance of the spark-ignition, ammonia-fueled engine decays rapidly at engine speeds above 3000 rpm. High speed output can be increased substantially by the addition of very small quantities (1.5 percent) of hydrogen.

6. Compression-ignition engines can operate with anhydrous ammonia as a fuel by simply supplying a means of introducing the ammonia into the induction air system. The existing fuel injection system can be used to supply a pilot charge of diesel fuel for initiating ignition.

7. To minimize the quantity of pilot fuel to acceptable levels requires changes to the fuel injection system that are not practical with today's commercial units.

8. The preferred means of operating a compression-ignition engine on anhydrous ammonia fuel is to convert it to a spark-ignition engine.

9. Because of their inherent high compression ratio and favorable air flow rates, diesel engines make excellent ammonia-fueled engines when converted to operate with spark-ignition.

10. Direct liquid injection of anhydrous ammonia is not practical in a compression-ignition engine.

11. Hydrogen is the only practical additive for use in remote areas as an ignition and flame propagation improver in a high speed ammonia engine.

12. Reassociation of radio-frequency (RF) dissociated nitrogen to provide ignition energy is not practical because of the short half-life of the dissociated products.

13. RF dissociation of ammonia as a source of hydrogen is not practical because of the high power requirements.

14. Anhydrous ammonia has no deleterious effects on engine components except for parts made of copper, brass or bronze.

## CONCLUSIONS

15. Ammonia is safe to handle, except in confined areas, because its low specific gravity causes it to rise in the atmosphere. Likewise ammonia in the exhaust of low-compression, spark-ignition engines resulting from partial burning at misfiring is not considered a problem.

## RECOMMENDATIONS

In order to fully explore the potential of anhydrous ammonia as a fuel for internal combustion engines, it is recommended that the development program be continued as follows:

1. That spark plugs with long electrodes and suitable for operation with high cylinder pressures be developed for improved durability.

2. That compact combustion chambers, approaching a spherical shape, be designed and tested for existing spark-ignition and compression-ignition engines.

3. That an ammonia vaporizer system utilizing exhaust gas energy be designed and developed.

4. That a dissociator system capable of producing 1.5 percent hydrogen by weight be designed and developed for topping off the high-load, high-speed end of spark-ignition engines having rated speeds above 3200 rpm.

5. That a spark-ignition engine suitable in design for operation at peak cylinder pressures of 1200 psi, or higher be tested using ammonia fuel.

## FACILITIES

In order to provide for safe handling and storage of anhydrous ammonia it was necessary to procure and install certain special equipment. A storage tank designed for 265 psi minimum pressure and having a capacity of 500 gallons,

FACILITIES

2400 pounds of ammonia, was obtained and installed on the roof above the two cells to be used. In addition to normal fittings and pressure relief valves, the tank was equipped with a 10 KW electric vaporizer with proper controls to maintain a tank pressure of 125 psi under all conditions of ambient temperature or rates of useage of ammonia. A water spray system and a metal awning to deflect direct sunshine were installed to prevent development of excessive pressures as a result of high temperatures. A remote reading liquid level indicator in the test cell control room showed the amount of fuel in the tank at all times.

The complete ammonia supply system was installed by a licensed contractor, Armour Agricultural Chemical Company. Prior to the first filling of the storage tank, the complete system was inspected and approved by the Department of Buildings and Safety Engineering of the City of Detroit in accordance with City Ordinances No. 518-E and 547-F.

Two separate fuel lines, one for liquid ammonia and one for ammonia vapor, were run between the storage tank and each test cell. In each of these lines were installed manual and solenoid operated shutoff valves, Cuno Micro-Klean filters, Rego pressure regulating valves, Cash air-operated control valves, and Brooks flowmeters. All equipment was designed specifically for use with ammonia. Figure 1 shows the schematic arrangement of the ammonia supply system.

General Electric motoring - absorption dynamometers with Link direct reading torquemeters were used for measuring engine output. Meriam laminar flow air meters and Meriam direct reading inclinometers were used for measuring induction air flow. The only special test equipment required for monitoring the engines (other than in the fuel system) were stainless steel Heise gauges. These were necessary because of the affinity of ammonia with mercury and copper.

Before installation each engine was modified to accept a Kistler Model 601-H water cooled, pressure pick-up. Output of the pressure pickups was amplified by a Kistler Model 504 charge amplifier and displayed on a Tektronix Model 502-A dual-beam osciliscope. Electro Model 3010-AN proximity pickups were used to trigger the osciliscope and for the display of timing marks.

The following safety equipment was available at all times in, or adjacent, to the control room:

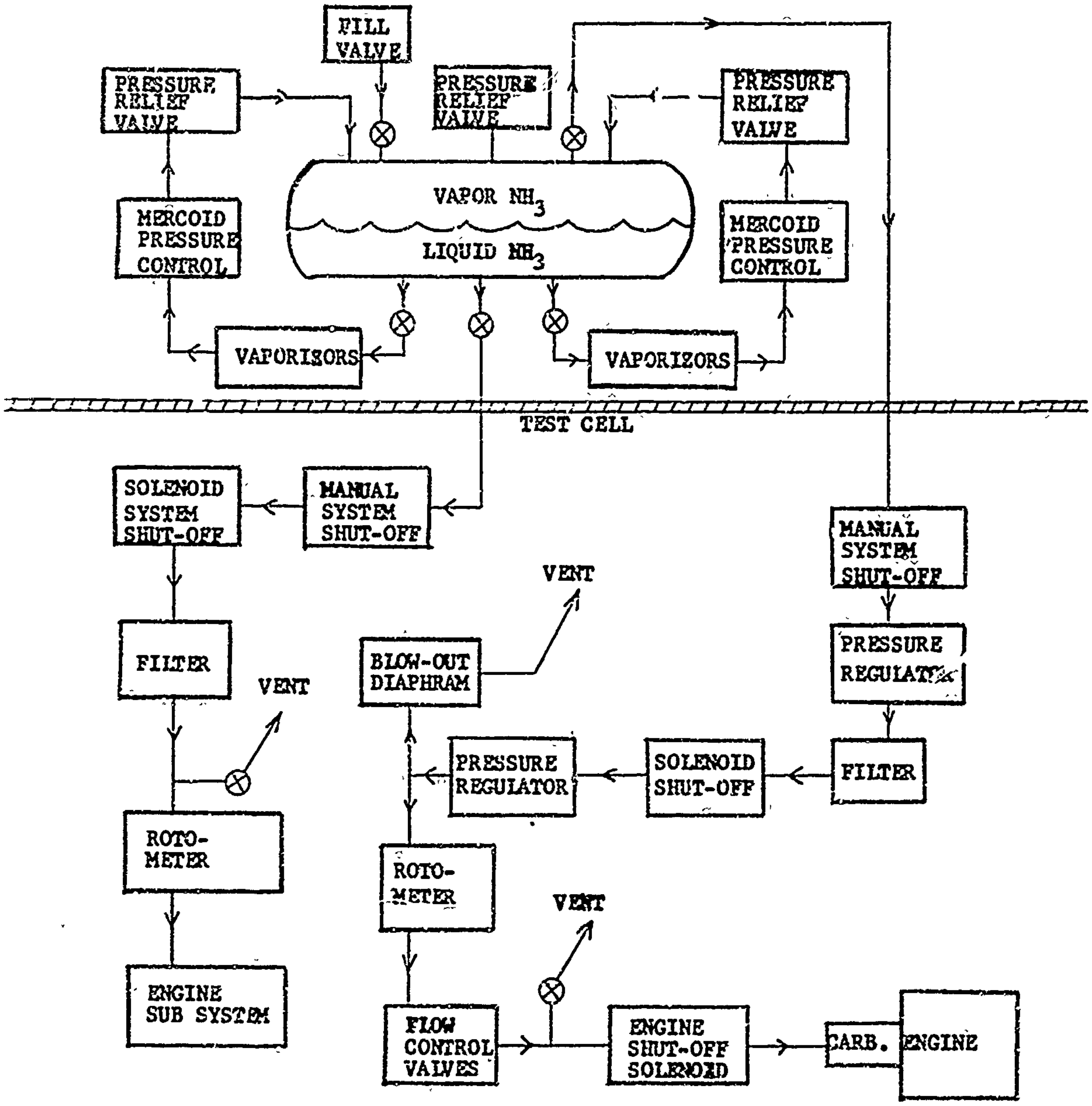



Fig. 1. Schematic Arrangement of Ammonia Supply System.

## FACILITIES

1. One quick opening, flooding type shower.
2. One quick opening, flooding type eye bath.
3. One Mine Safety Appliances (MSA) oxygen breathing apparatus.
4. Four MSA canister ammonia gas masks.
5. First aid kit.
6. Vapor proof goggles.
7. Rubber gloves.

Figures 2 and 3 show the common control room between the two test cells. Figure 4 shows the L-141 engine as installed in test cell A-8. Figure 5 shows the AVDS-1790 Vee Twin as installed in test cell A-7.

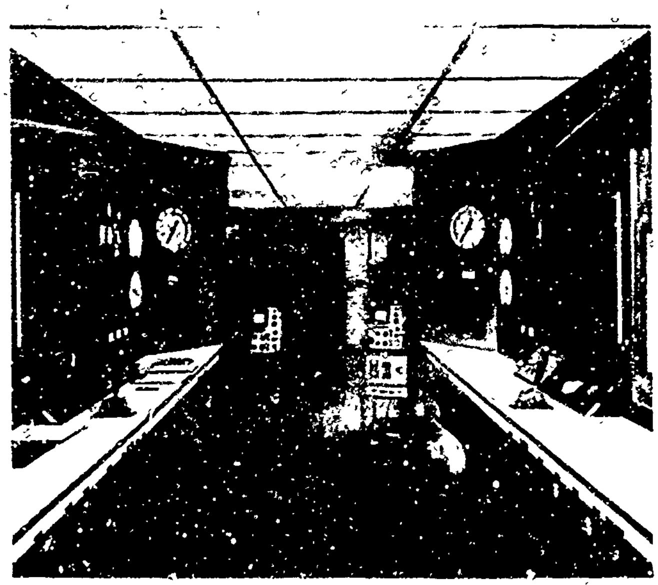


Fig. 2. Ammonia Test Facility Control Room. (D-33751)

Fig. 3. Control Panel of One of the Ammonia Test Cells. (D-33753)

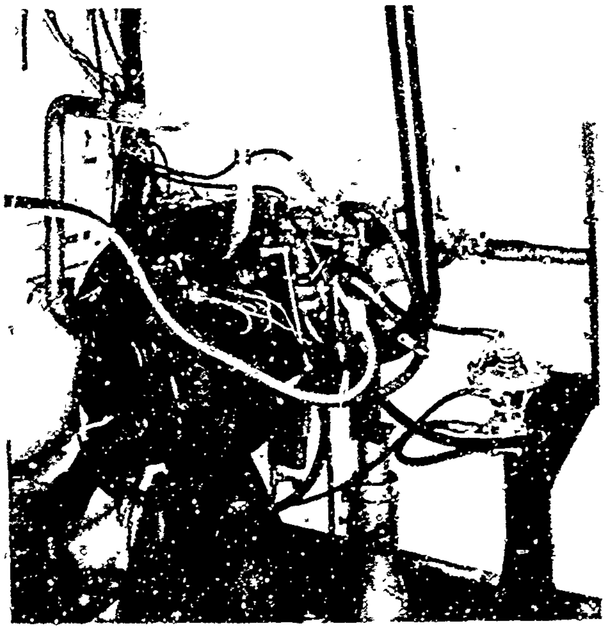


Fig. 4. L-141 Engine As Installed For Operation on Ammonia Fuel. (D-33500)

Fig. 5. AVDS-1790 Vee Twin As Installed For Operation on Ammonia Fuel. (D-35665)

## DISCUSSION

### SPARK-IGNITION ENGINE INVESTIGATION

An L-141 engine, as received from the Government, was installed in Continental test cell number A-8. It was given a shake-down run on gasoline and then was operated to obtain baseline fuel consumption at part load and various speeds. A total of three L-141 engines were used under this contract and they had a large variation in brake specific fuel consumption. Figures 6 through 13 show the envelope of fuel consumption versus brake mean effective pressure at speeds of 1200 to 4000 rpm for the three engines.

The first step in adapting this engine to run on ammonia was simply to install an LPG type carburetor. With no other changes, the engine was able to run but performance was very erratic; maximum speed obtained was 1045 rpm with the engine developing 0.895 horsepower. Subsequent changes to the carburetor chok.., venturi and main metering jet permitted the engine to produce a maximum of 9.2 horsepower at 1200 rpm and a maximum speed

DISCUSSION

of 1951 rpm. From this humble beginning many changes were made permitting dramatic improvements in performance. Unfortunately, as combustion of ammonia improved, peak firing pressures increased beyond the safe limit for operation of the cast iron crankshaft.

A torsional survey was run on the L-141 engine and dynamometer system. No serious torsional vibration frequencies exist in the system as installed and operated at Continental. An L-141 engine crankshaft was statically loaded in torsion and bending. The results of this investigation showed peak cylinder pressures should be limited to 900 psi. For this reason, the results shown in this report will not always be the optimum that could be obtained in an engine designed for operation on ammonia, likewise, the results reported will not be in chronological order, but will be grouped to show trends of the varous variables investigated..

Compression Ratio

Figure 14 shows the large improvement in performance that can be obtained by increasing compression ratio, particularly at high speeds. Additional work done on the Vee-Twin diesel engine, and discussed later in this report, indicates that the optimum compression ratio for spark-ignition engines burning ammonia should be about 16:1. This is believed to be due to the great increase in flame propagation rates as compression ratio is increased as reported by Samuelsen, Reference 2.

Figure 15 illustrates the effect of compression ratio on peak power and peak cylinder pressure. While cylinder peak pressure appears to have a direct linear relationship to compression ratio, the output is improving at a decreasing rate with increases in compression ratio.

Ignition Systems

It has always been known that ammonia would be difficult to ignite. Buckley and Husa reported in 1962, Reference 3, that the minimum ignition energy was 680 millijoules. Verkamp et al, Reference 4, have shown the ignition energy required is only eight millijoules. Verkamp has further shown that relatively small amounts of hydrogen reduce the required energy level to that of hydrocarbons (0.3 millijoules). Figure 16 is a replot of Verkamps's data showing the ignition energy of ammonia and ammonia plus hydrogen versus equivalence ratio.

NH3- 209

Composite Part Load Curves at 1200 RPM For Engine Nos. 5616, 5904, and 5677 on 80 Octane Fuel (MIL-G-3065)

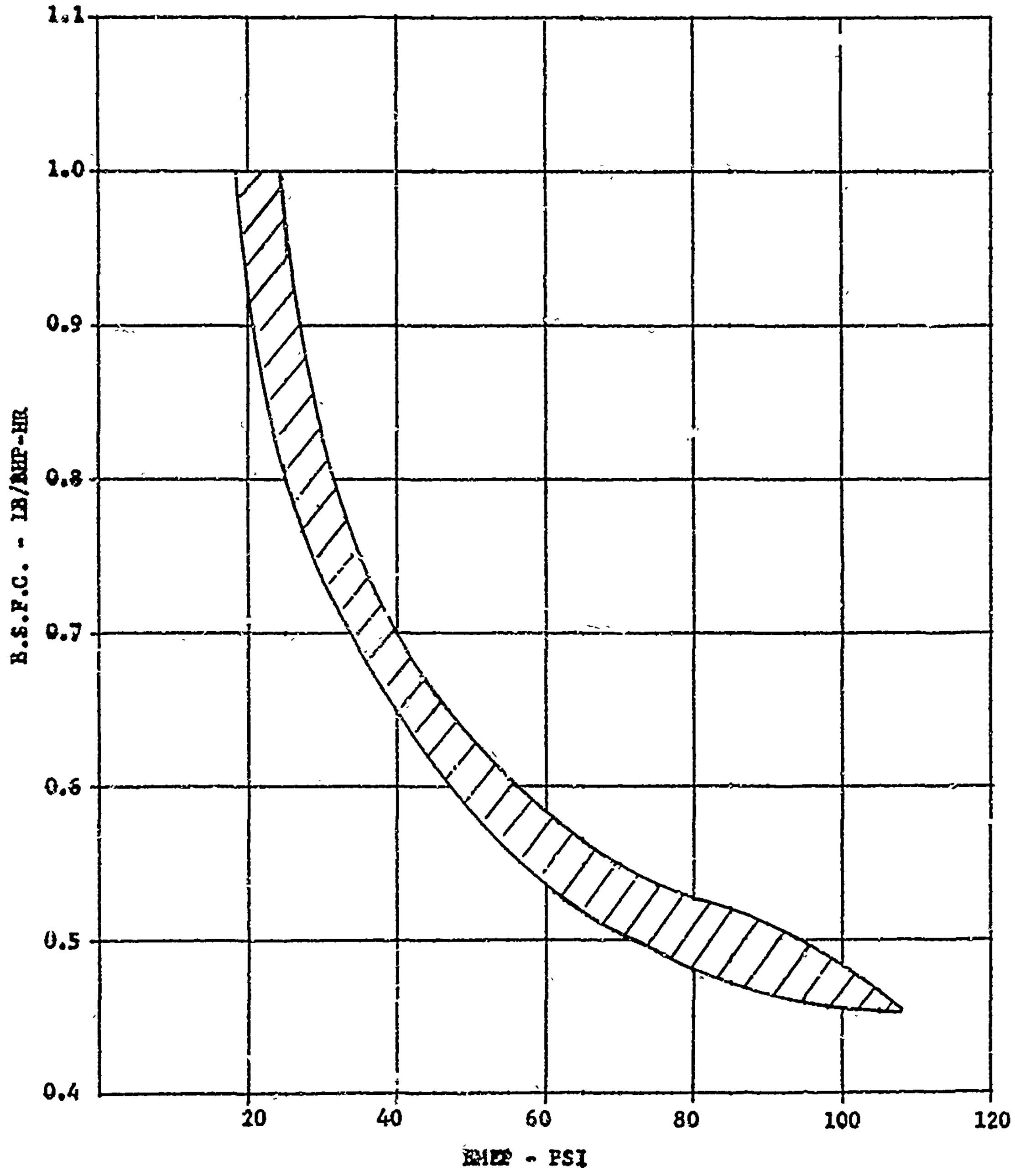





Fig. 6

Composite Part Lead Curves At 1600 RPM for Engine Nos. 5616, 5904 and 5677 on 80 Octane Fuel (MIL-G-3065)

J. Pinter

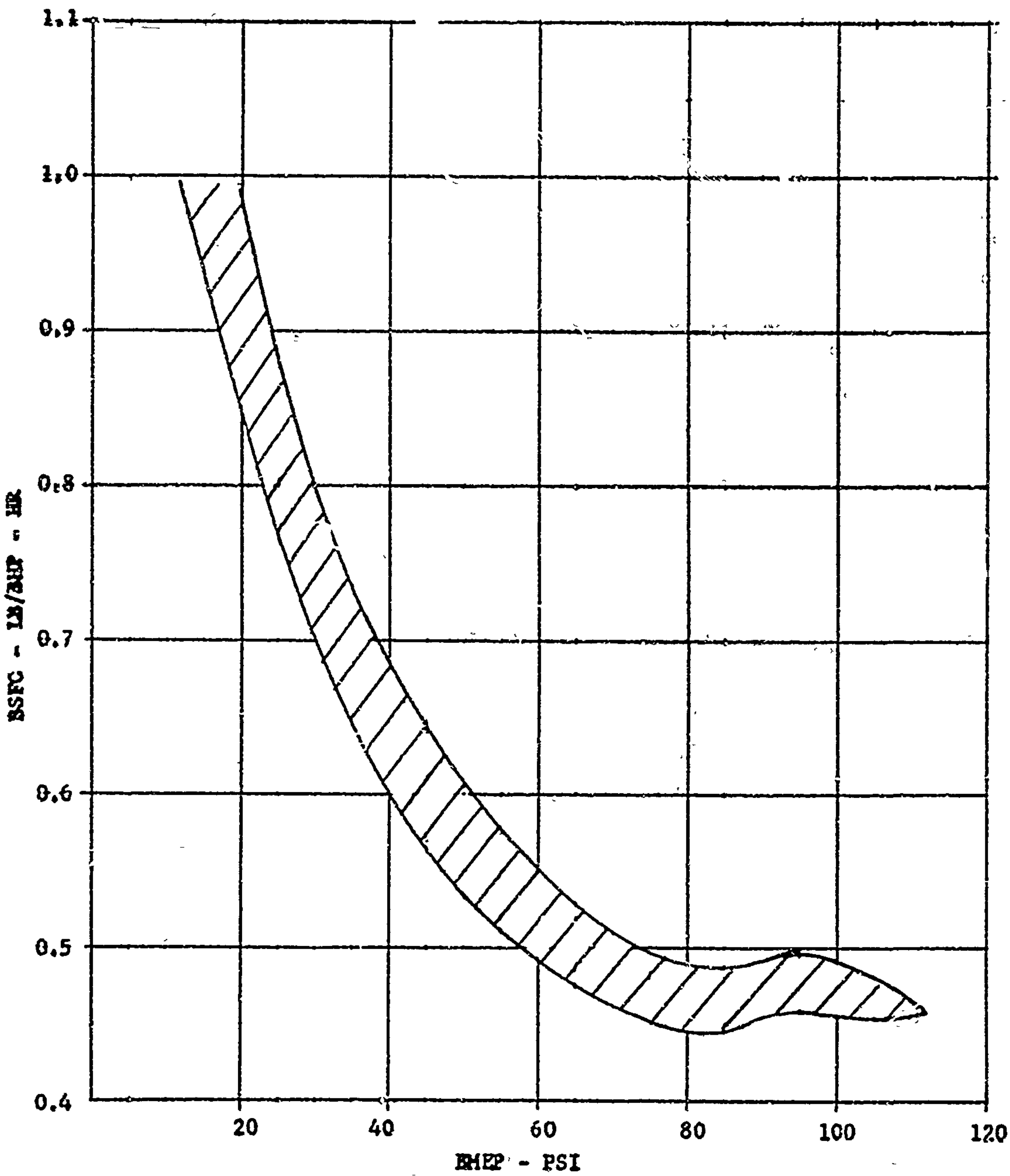



Fig. 7

$\overline{NH_3}$- 211

. Composite Part Load Curves at 2000 RPM for Engine Nos. 5616, 5904, and 5677 on 80 Octane Fuel (MIL-G-3065)

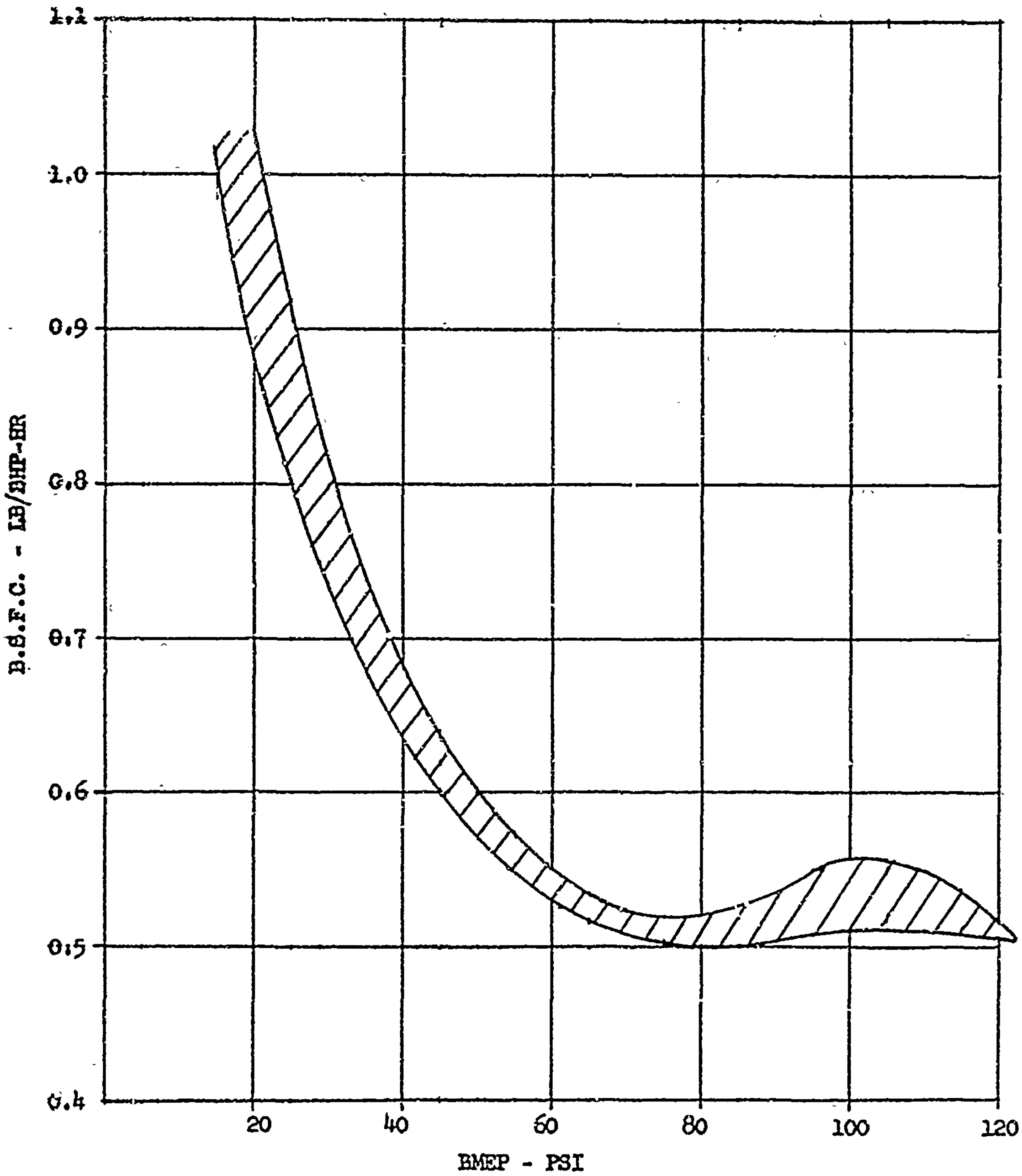





Fig. 8

Composite Part Load Curves at 2400 RPM For Engine Nos. 5616, 5904 and 5677 on 80 Octane Fuel (MIL-G-3065)

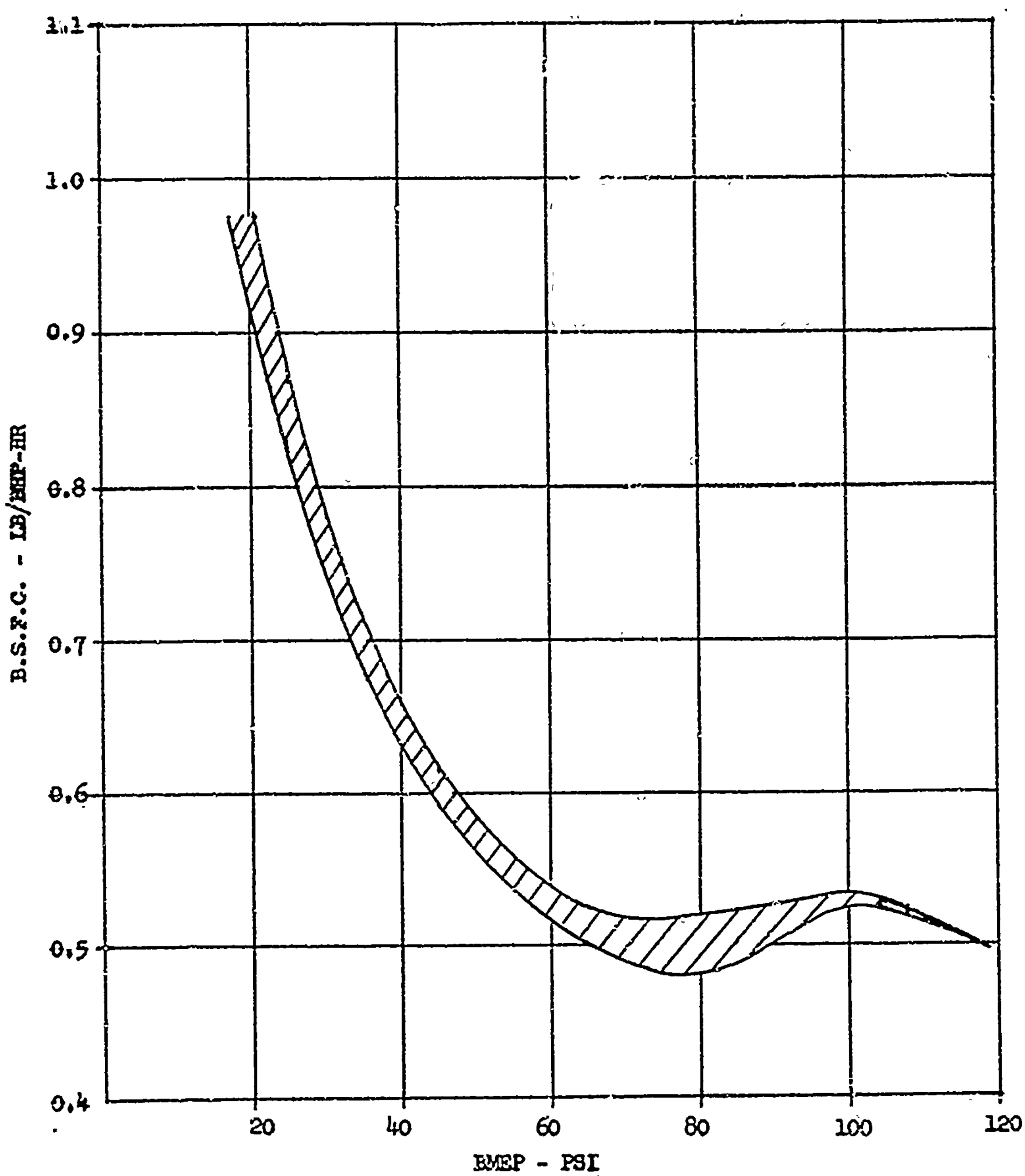



Fig. 9

$NH_3$ - 213

L-141

Composite Part Load Curves at 2800 RPM For Engine Nos. 5616, 5904, and 5677
80 Octane Fuel (MIL-G-3065)

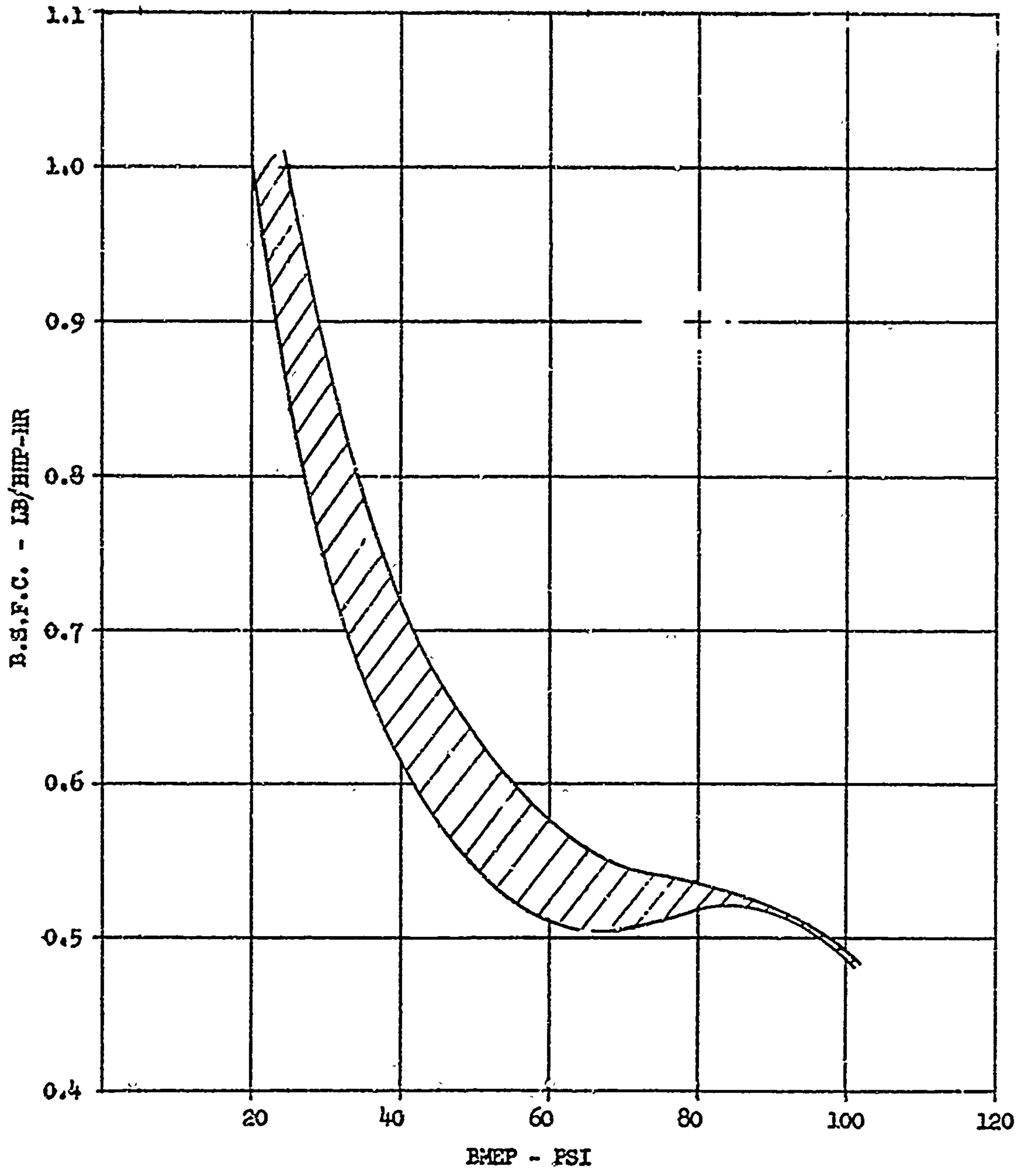





Fig. 10

L-141

Composite Part Load Curves at 3200 RPM For Engine Nos. 5616, 5904, and 5677 on 80 Octane Fuel (MIL-G-3065)

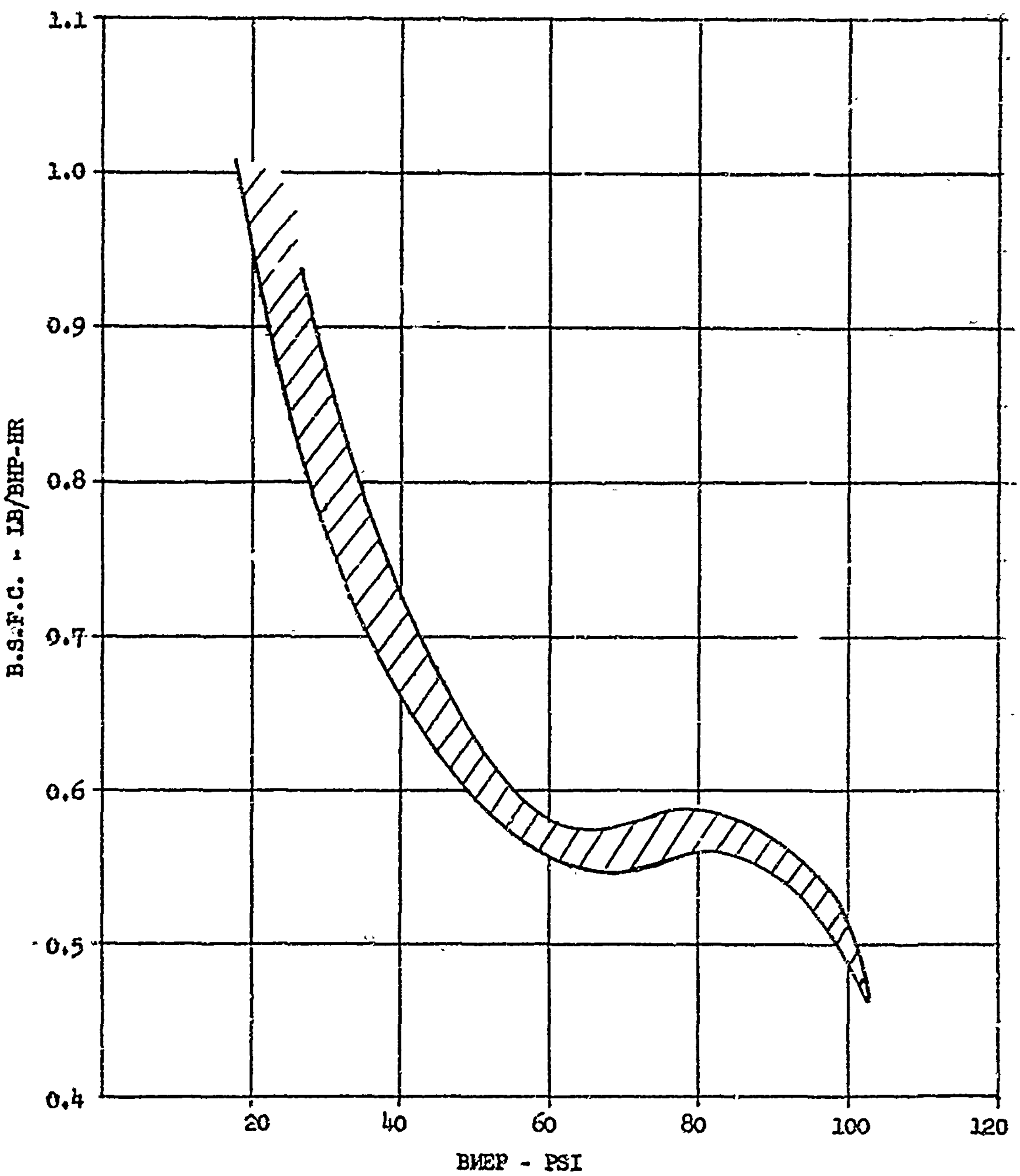



Fig. 11

NH3 - 215

L-141

Composite Part Load Curves at 3600 RPM For Engine Nos. 5616, 5904, and 5677 on 80 Octane Fuel (MIL-G-3065)

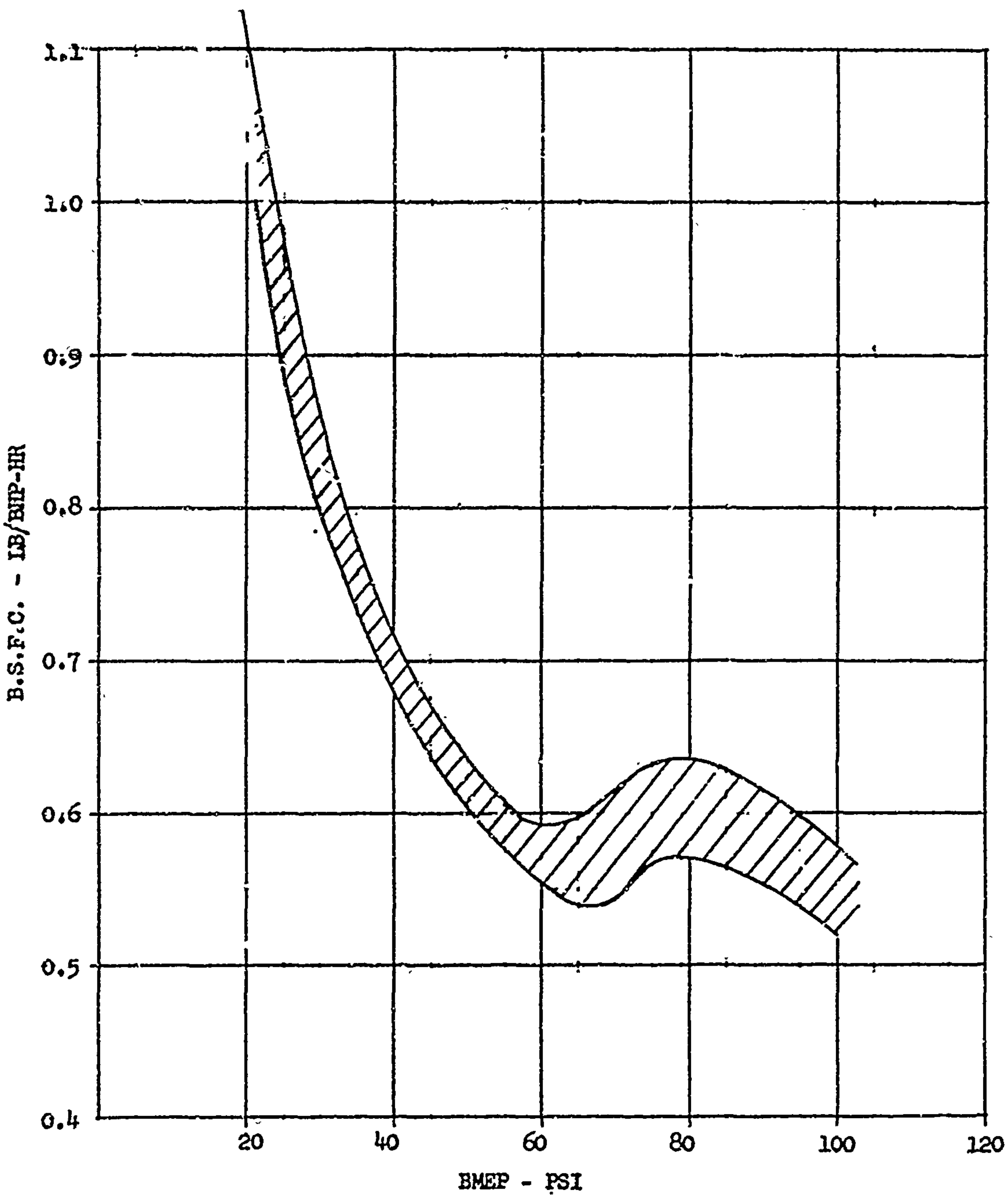



Fig. 12

Composite Part Load Curves at 4000 RPM for Engines Nos. 5616, 5904 and 5677 on 80 Octane Fuel (MIL-G-3065)

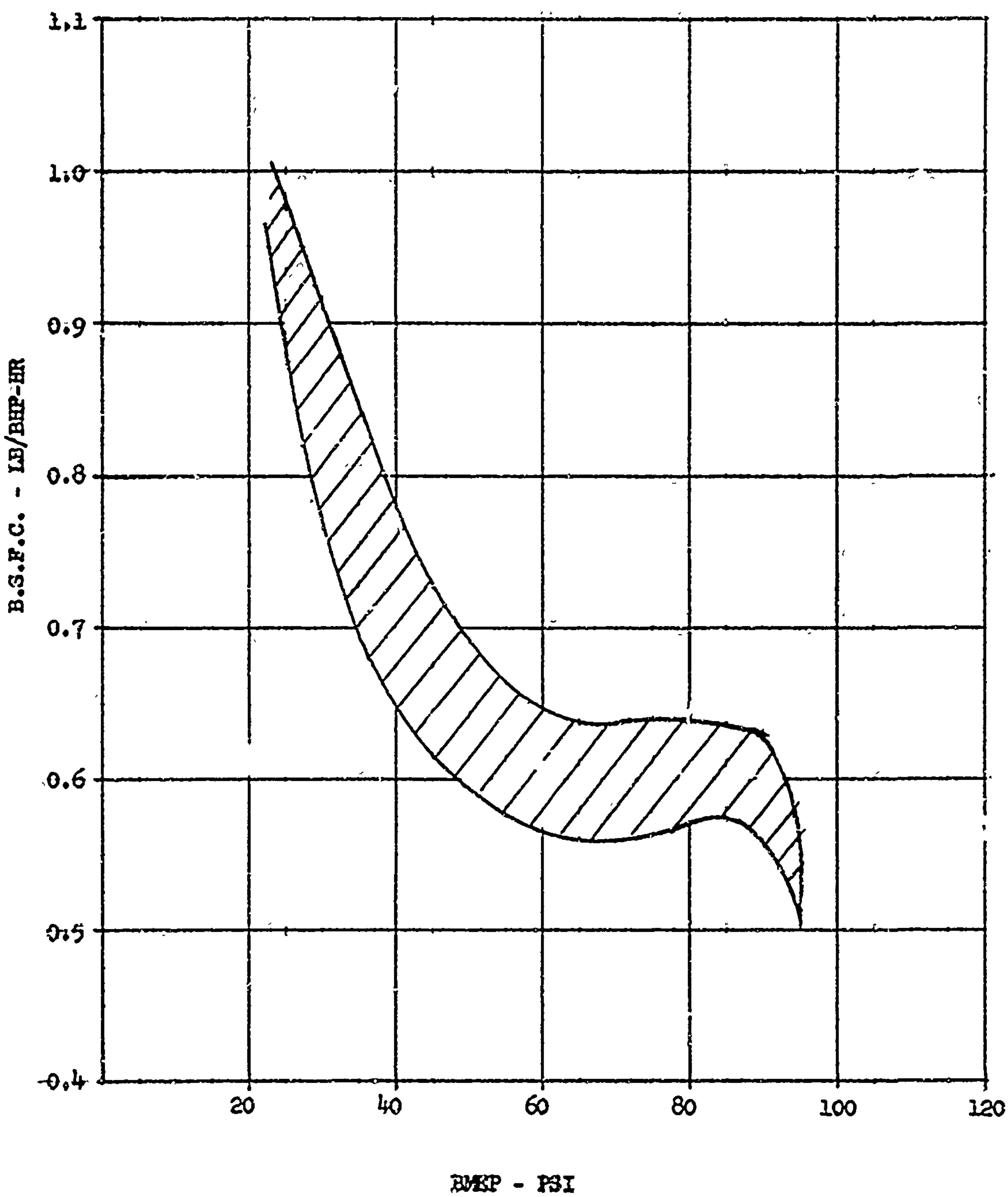



Fig. 13

Effect of Compression Ratio on Performance (Champion N-11y Spark Plugs, 0.100 in. Gap, Mallory Super Magneto)

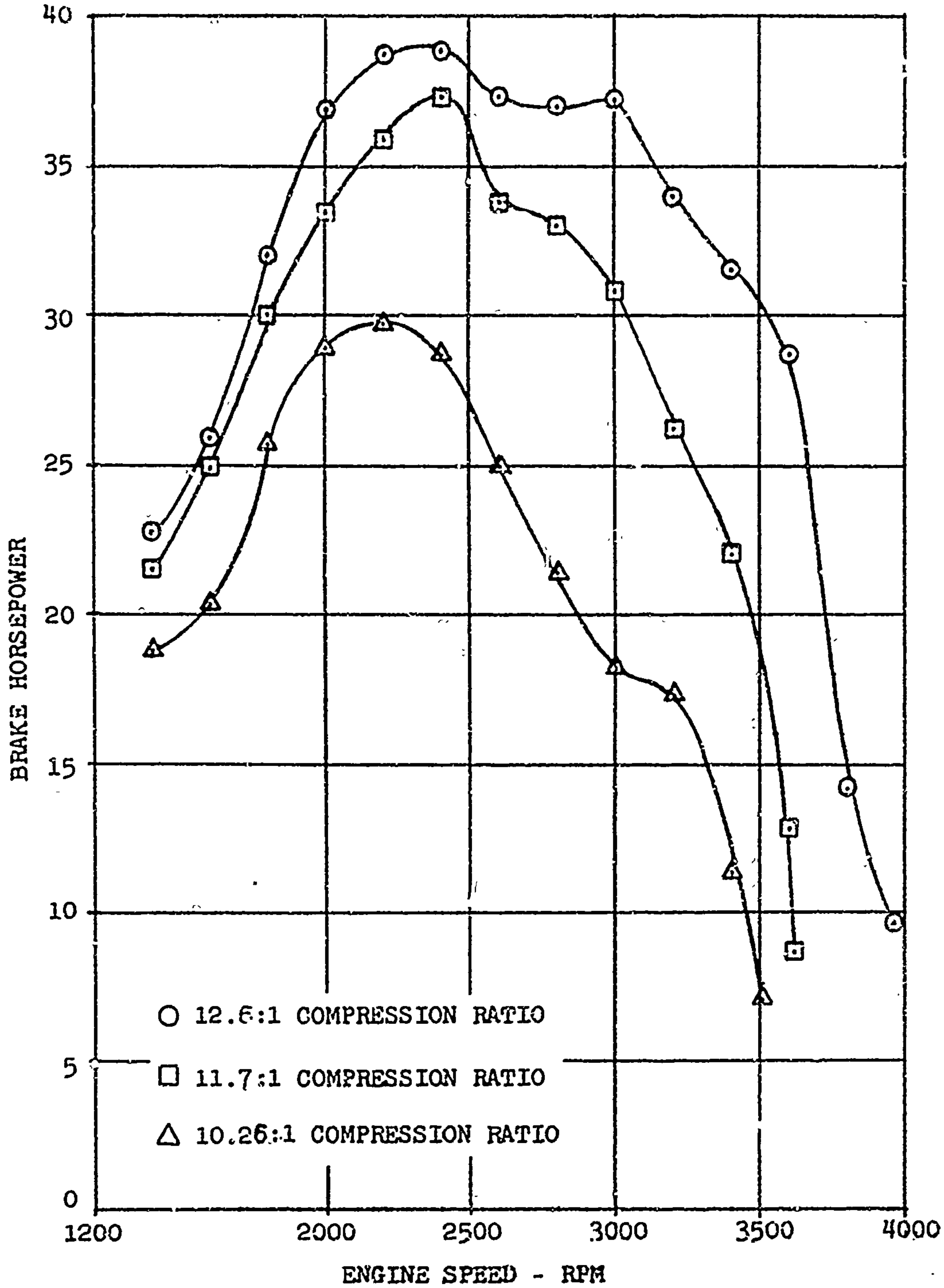



Fig. 14

Effect of Compression Ratio on Peak Cylinder Pressures (Champion N-11y Plugs, 0.100 in. Gap, Mallory Magneto)

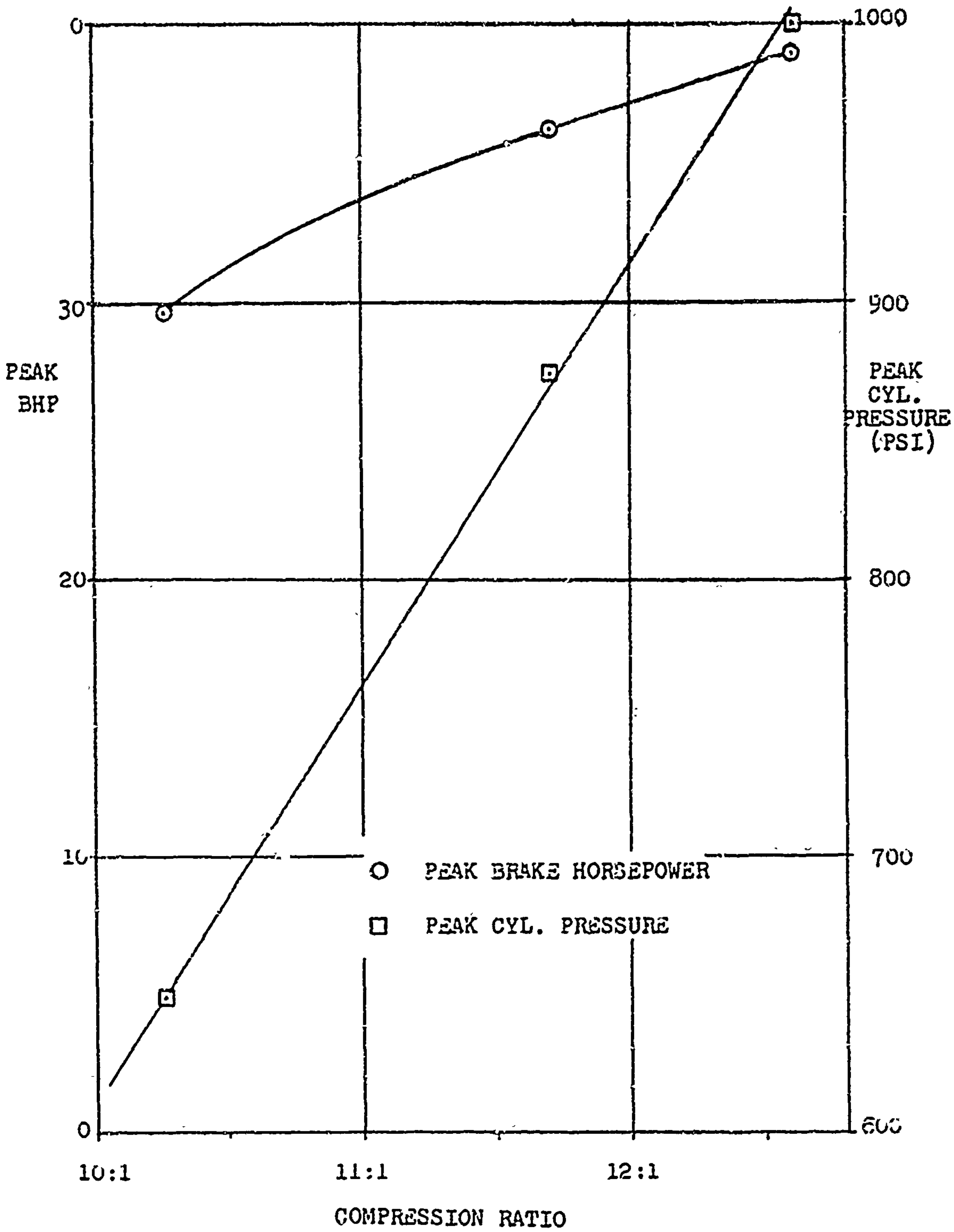





Fig. 15

IGNITION ENERGY OF AMMONIA PLUS HYDROGEN

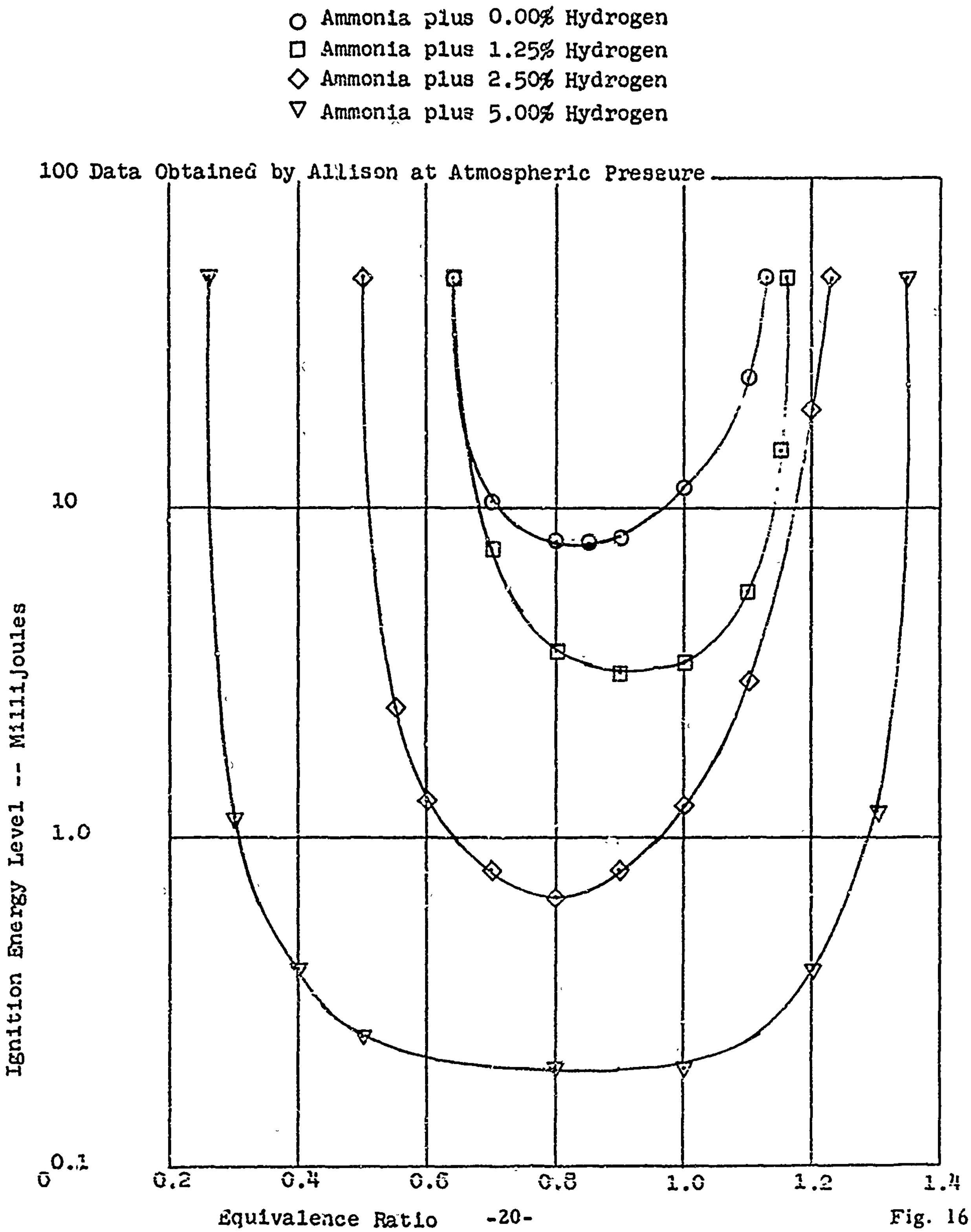





Fig. 16

## DISCUSSION

A total of six ignition systems, over and above standard engine equipment, were procured for evaluation. These were:

1. Prestolite capacitor discharge system.
2. Prestolite Transitorized Ignition System.
3. A Mallory "Hot Rod" Coil, Model U12A.
4. Texaco Continuous Arcing System.
5. Mallory "Super Mag" Magneto.
6. Mallory Capacitor Discharge System.

Each of these sources of ignition energy, except the Capacity Discharge Systems, produced a substantial increase both in peak power obtainable and maximum permissible operating speed. The Mallory Magneto, however, was unquestionably the best at all speeds above 1400 rpm. It appears that ammonia fuel wants a "fat", "hot" spark with long duration. All of the above induction type systems had a spark duration of about 0.5 milliseconds whereas the capacitor discharge system had a spark duration of only 0.02 to 0.07 milliseconds. The following tabulation illustrates the characteristics of the various ignition systems tested when operated with a 100 picofarad load.

| Ignition System | KV Output | Energy Output Millijoules | Spark Duration Milliseconds |
|---|---|---|---|
| Prestolite C.D. | 28 | 41.5 | 0.07 |
| Prestolite Transitorized | 23 | 29 | 0.5 |
| Mallory Coil | 24 | 31.1 | 0.4 |
| Texaco | 20 | 21.9 | --- |
| Mallory Magneto | 31 | 51.9 | 0.5 |
| Mallory C.D. | 27 | 40 | 0.02 |
| Standard L-141 Coil | 15 | 13 | 0.5 |

Figure 17 is a plot of engine output versus speed for each of the various ignition systems. It is noted that the inductance systems, even with lower energy

NH3-219

Ignition System Performance on Ammonia Fuel, 12.6:1 Compression Ratio and Standard Spark Plugs with 0.100 in. Gap.

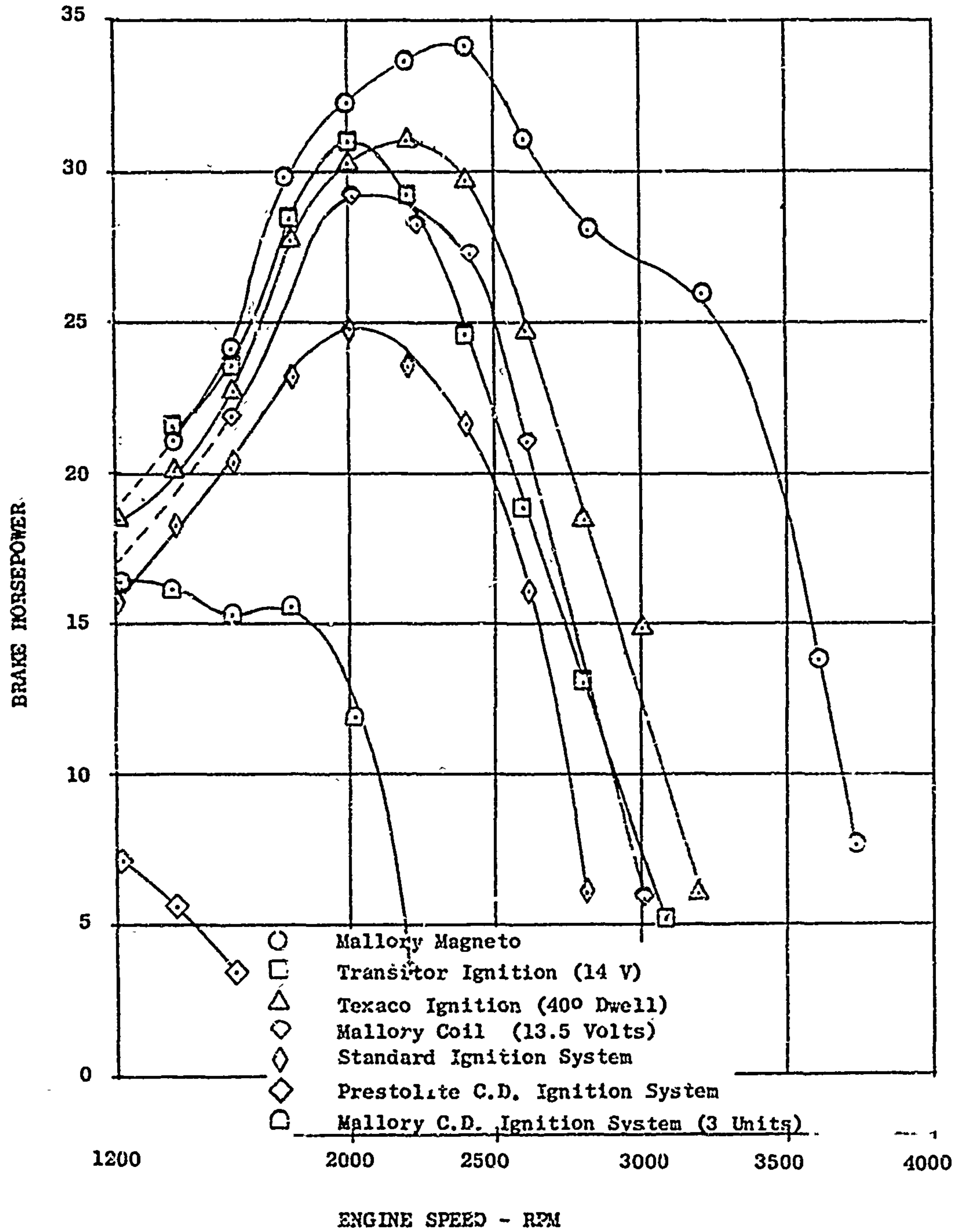





Fig. 17

DISCUSSION

outputs are clearly superior to the two capacitance systems tested. Figure 18 shows the effect of ignition energy on power output, for the various ignition systems tested.

## Spark-Plug Gap

Increasing the spark plug gap from 0.030 to 0.100 inch with the standard L-141 ignition system and 12.6:1 compression ratio resulted in an increase in maximum power developed from 19.6 to 27.0 horsepower. Maximum engine speed increased from 2400 to 3100 rpm. Figure 19 shows the effect of further increases in spark plug gap at a somewhat lower (11.7:1) compression ratio. It is noted that the effect is much greater at 4000 rpm than at lower speeds and also that from 0.110 to 0.120 inch the improvement is getting asymptotic. Another interesting phenomena was that, depending upon the type of spark plug used, if the gap is increased beyond the clearance between the center electrode and the body of the plug - there was a severe deterioration in performance. This is apparently because the plug sparks between the center electrode and the outer wall with a multiplicity of small sparks, rather than one "fat" spark at the gap.

## Spark Plug Reach

The standard military spark plugs having center electrodes practically flush with the cylinder head were replaced with champion N-11Y spark plugs having electrodes that extended into the combustion chamber an additional one-half inch. A dramatic improvement in performance was observed. Still longer reach plugs, champion OJ-22-1, were tried with additional gains in performance. Each type of spark plug was tested with spacers and modifications to the electrodes to cover locations from one side of the combustion chamber to the opposite side. The optimum location was determined to be 0.875 inch into the combustion chamber, which is approximately the center of mass of the combustion chamber at top dead center and as low as the electrodes can go without being struck by the pistons. Figures 20 and 21 show the spark plugs as installed. Figure 22 shows the effect of spark plug reach on peak horsepower output.

Since these plugs were designed for igniter service, not engine operation, they have limited line. Figure 23 shows the effect of overheating of the electrodes after 122 hours of operation. The spark plug manufacturers have stated that satisfactory plugs could be developed for this purpose.

Effect of Ignition Energy on Power Output; Ammonia Vapor Fuel -- 12.6:1 C.R.--
.05 Milli Sec. Spark Duration
T.J. Pearsall

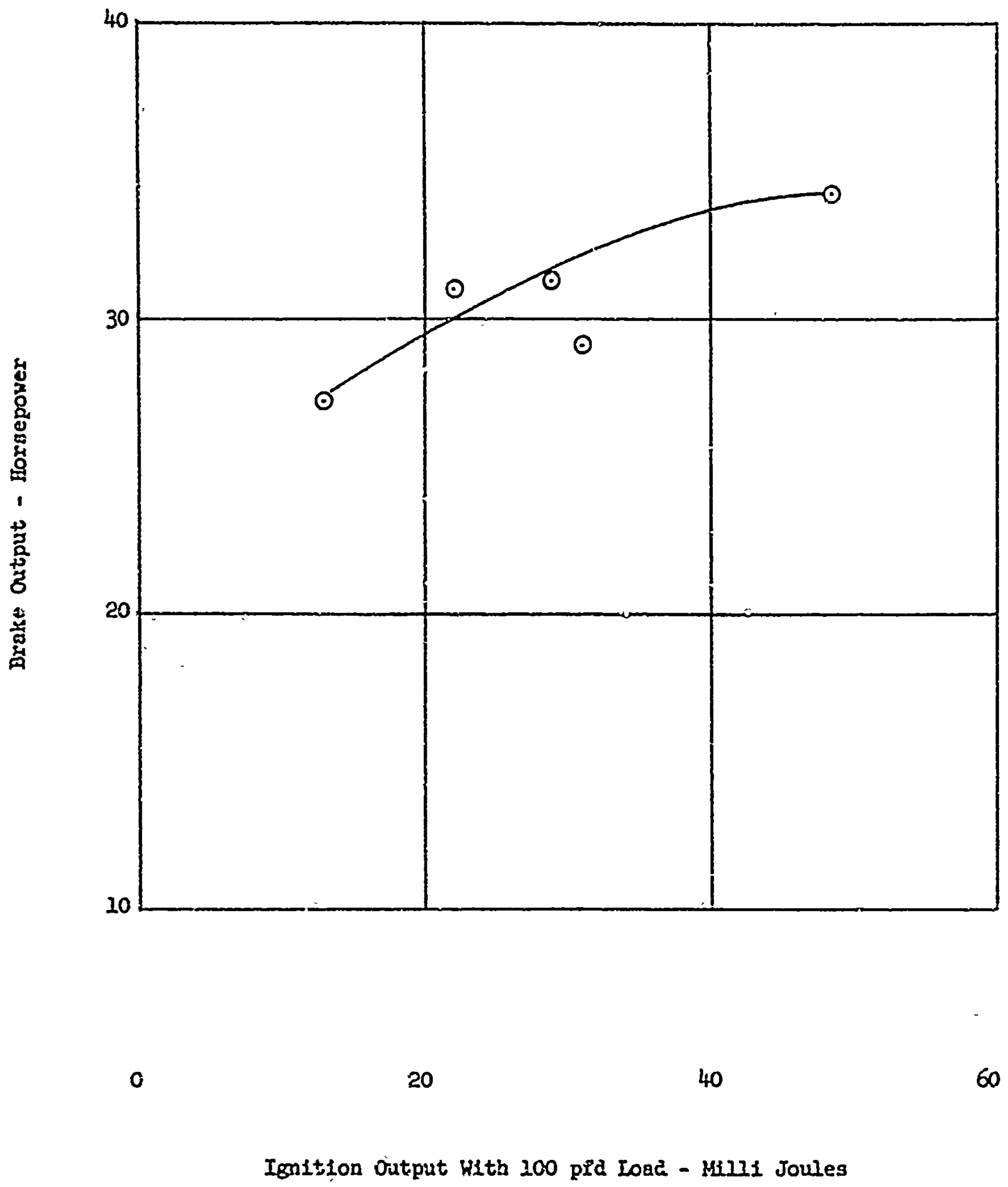



Fig. 18

$NH_3$ - 192

L-141

Effect of Spark Plug Gap on Brake Horsepower (Champion OJ-22-1 Spark Plugs, Mallory Super Magneto, 11.7:1 Compression Ratio)

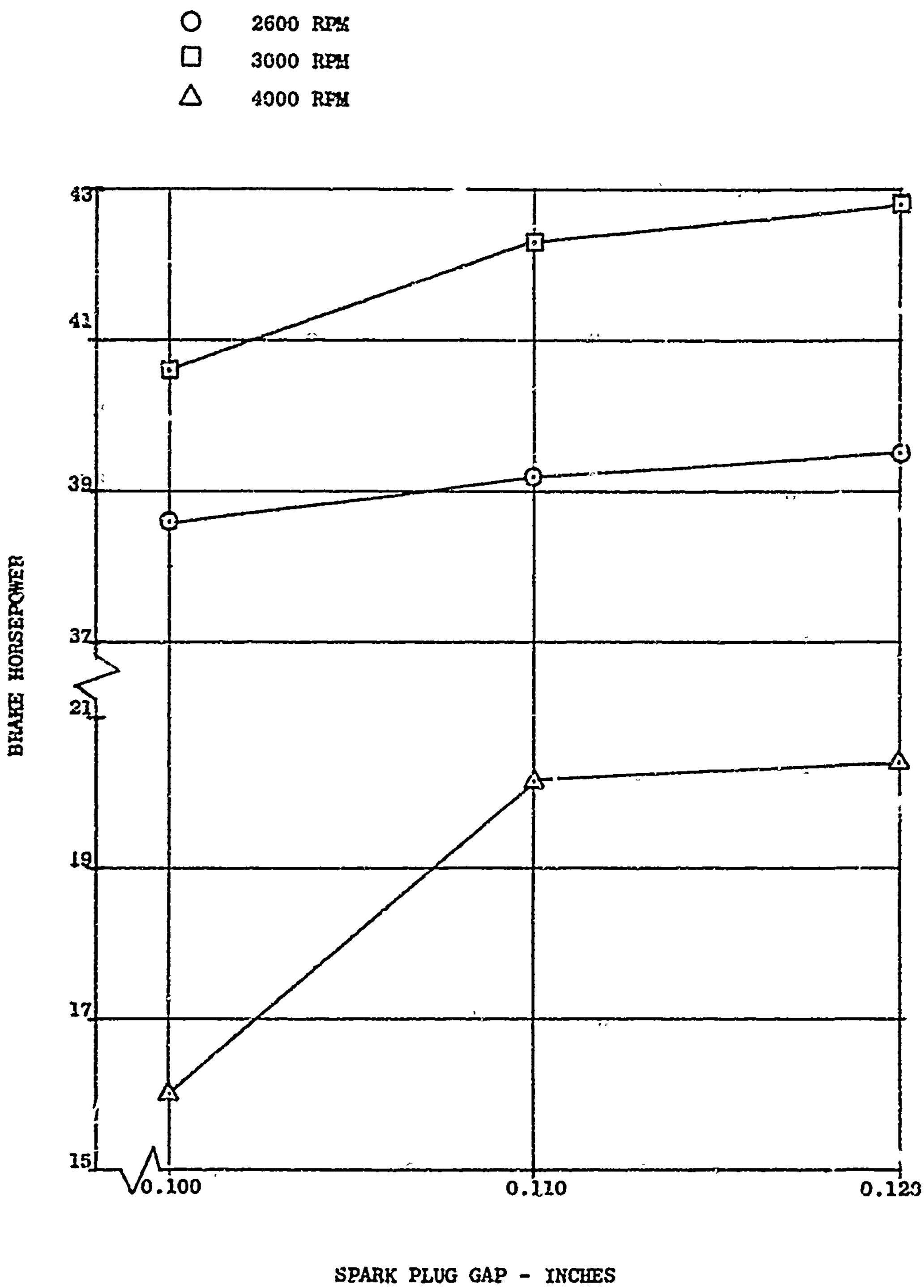





Fig. 19

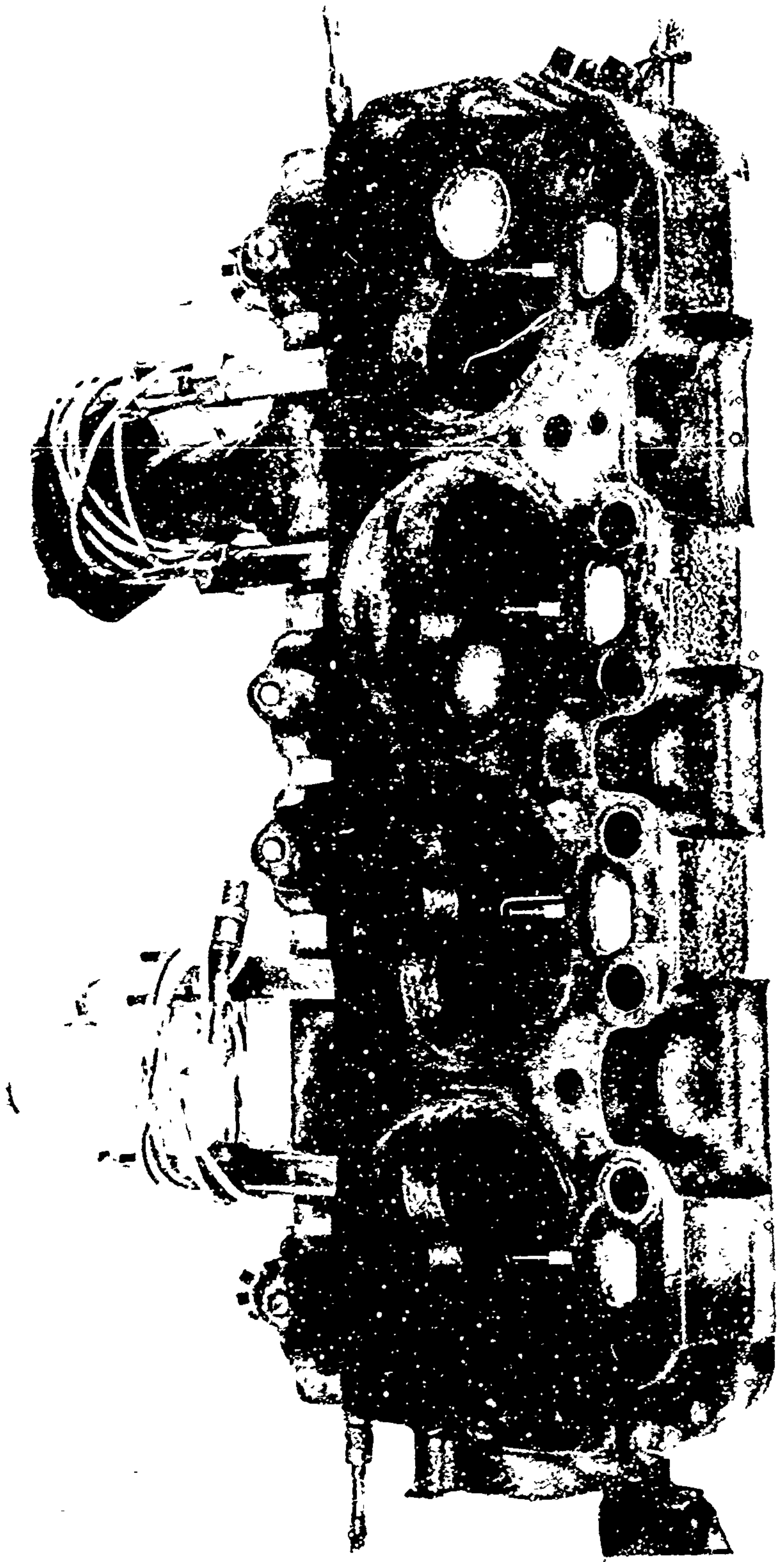


Fig. 20. Installation of Champion OJ-22-1 Spark Plugs In L-141 Cylinder Head. (D-35588)

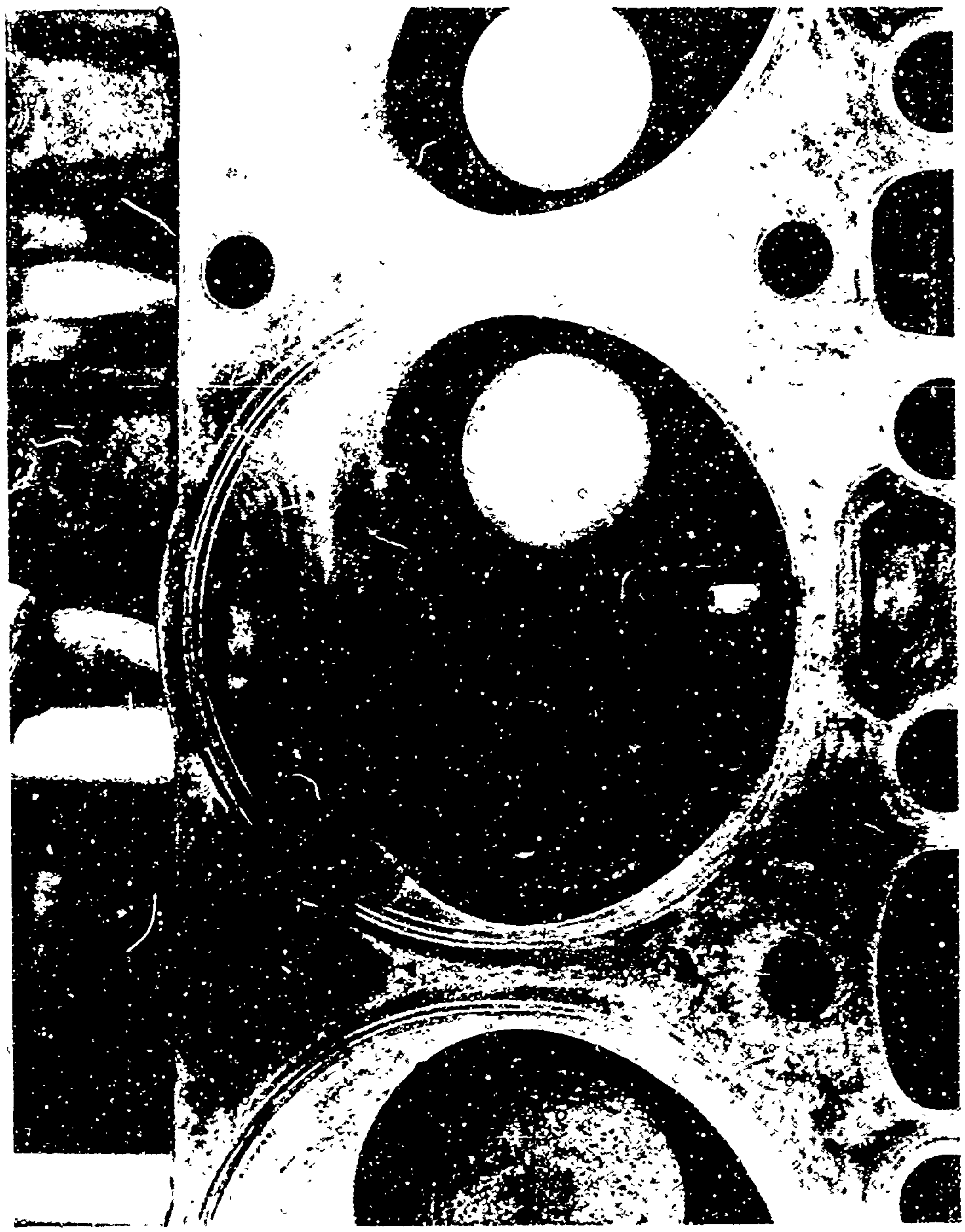


Fig. 21. Close Up View of Spark Plug Installation. (D-35589)

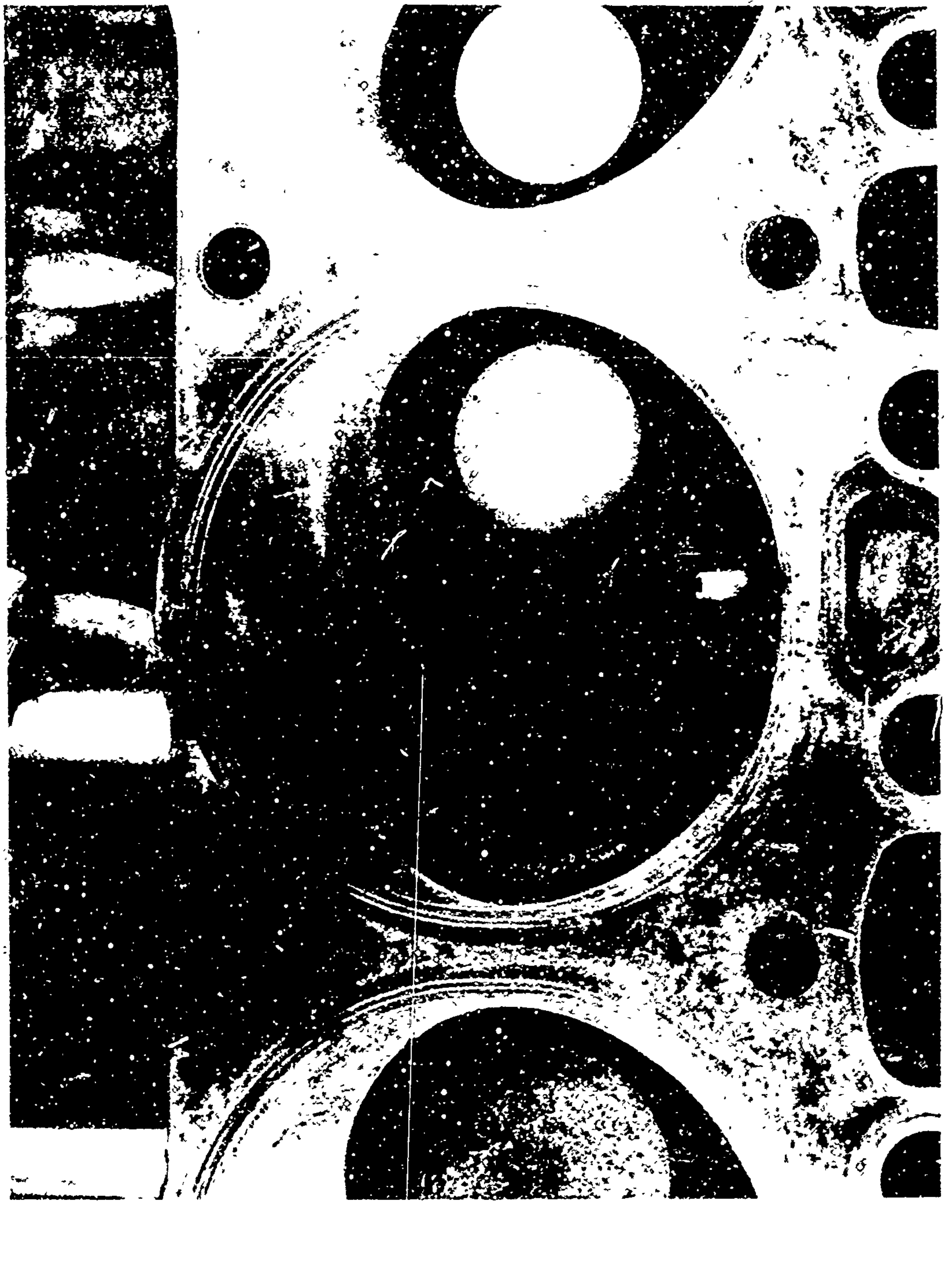


Fig. 21. Close Up View of Spark Plug Installation. (D-35589)

DISCUSSION

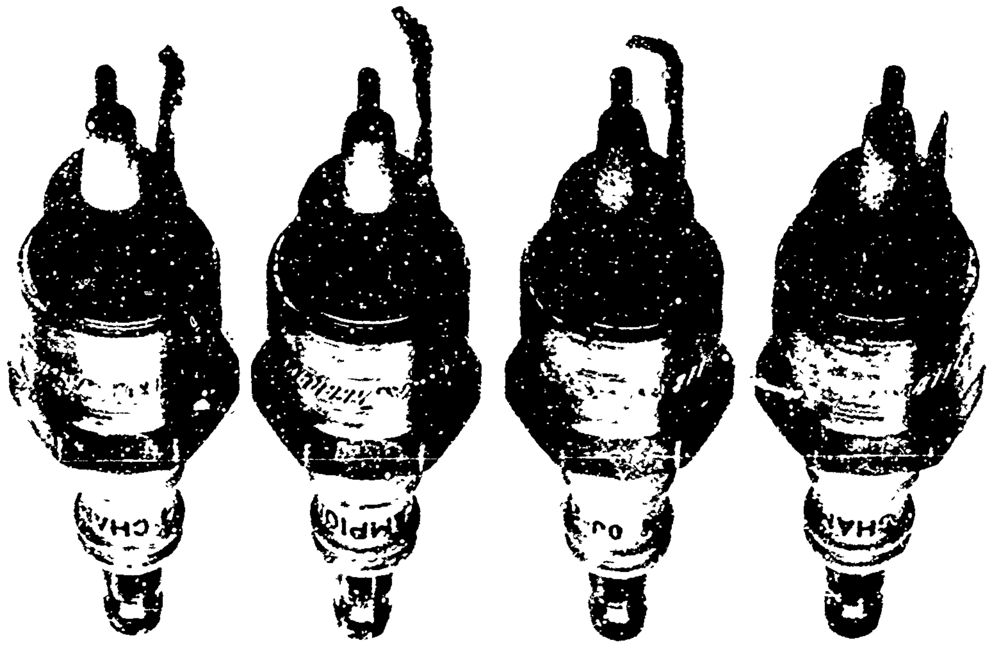


Fig. 23. Used Spark Plugs After 122 Hours of Operation With Ammonia Fuel. (D-35195)

## Fuel Supply System

As originally set-up, the L-141 engine was equipped with an LPG carburetor modified for use with ammonia. The modifications were of a minimum nature, primarily to eliminate parts that would be attacked by ammonia and to increase the fuel flow. Additional modifications were made as testing progressed to the point where the carburetor no longer acted as a fuel measuring device but as a mixing chamber. Based on actual engine performance a carburetor specification was developed and hardware procured. Figures 24, 25, and 26 show the ammonia flow, hydrogen flow and air flow requirements of the engine versus rpm. Figure 27 shows a comparison in performance with the carburetor and with the mixing chamber showing the excellent match between desired and achieved fuel flow.

## Swirl

Originally the engine was operated with the air induction system as received. It was considered, however, that the poor combustion characteristics of ammonia could be improved by inducing some swirl to the incoming air-fuel

NH3-173

L-141

Ammonia flow Vs. RPM (Magneto Ignition, 12.6:1 C.R. Head, Optimum Fuel Flows and Timing)

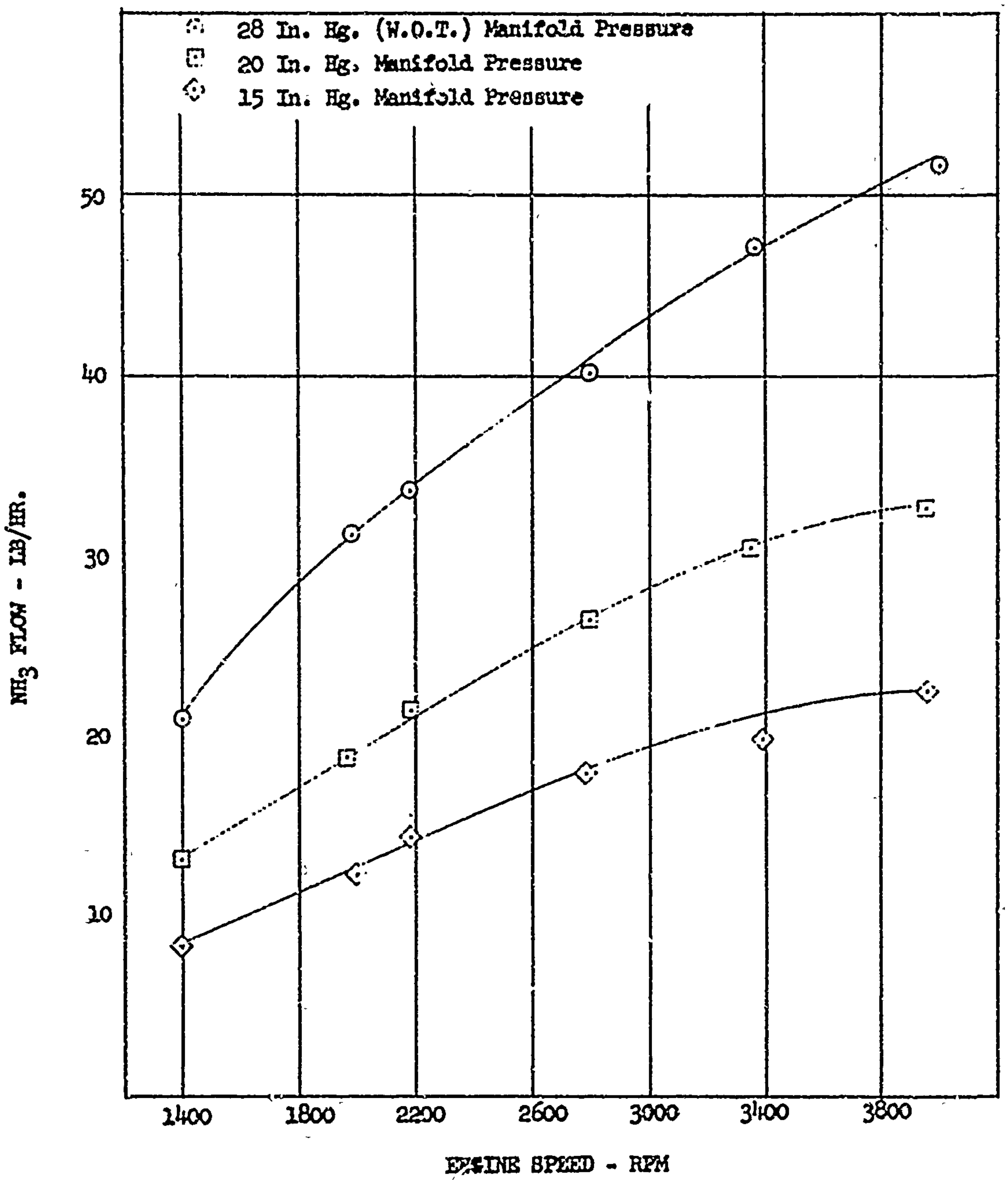





Fig. 24

L-141

Hydrogen Flow Vs. RPM (Magneto Ignition, 12.6:1 C.R. Head, Optimum Fuel Flows and Timing)

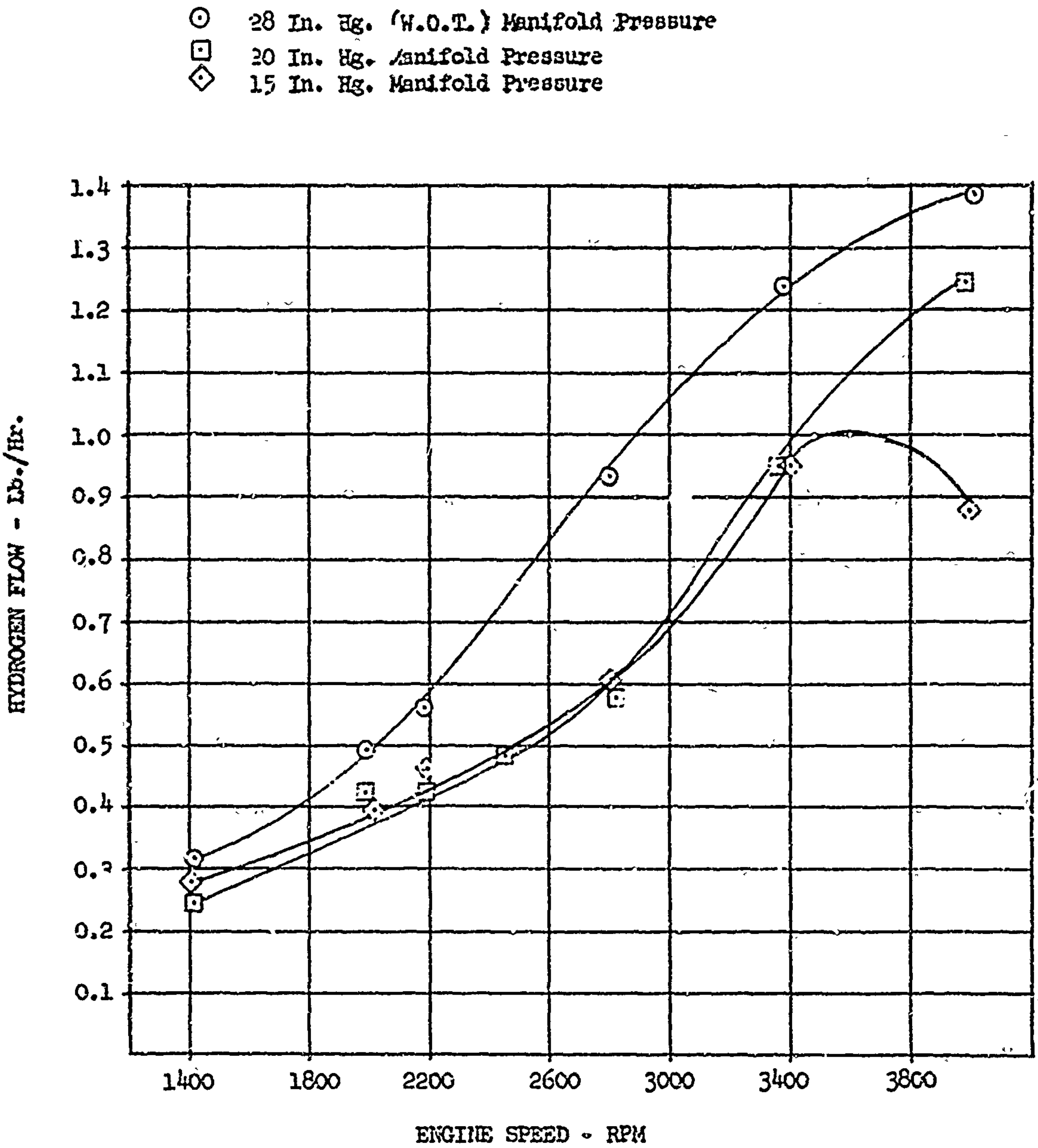





Fig. 25

Airflows Vs. RPM (Magneto Ignition, 12.6:1 C.R. Head, Optimum Fuel Flows and Timing)

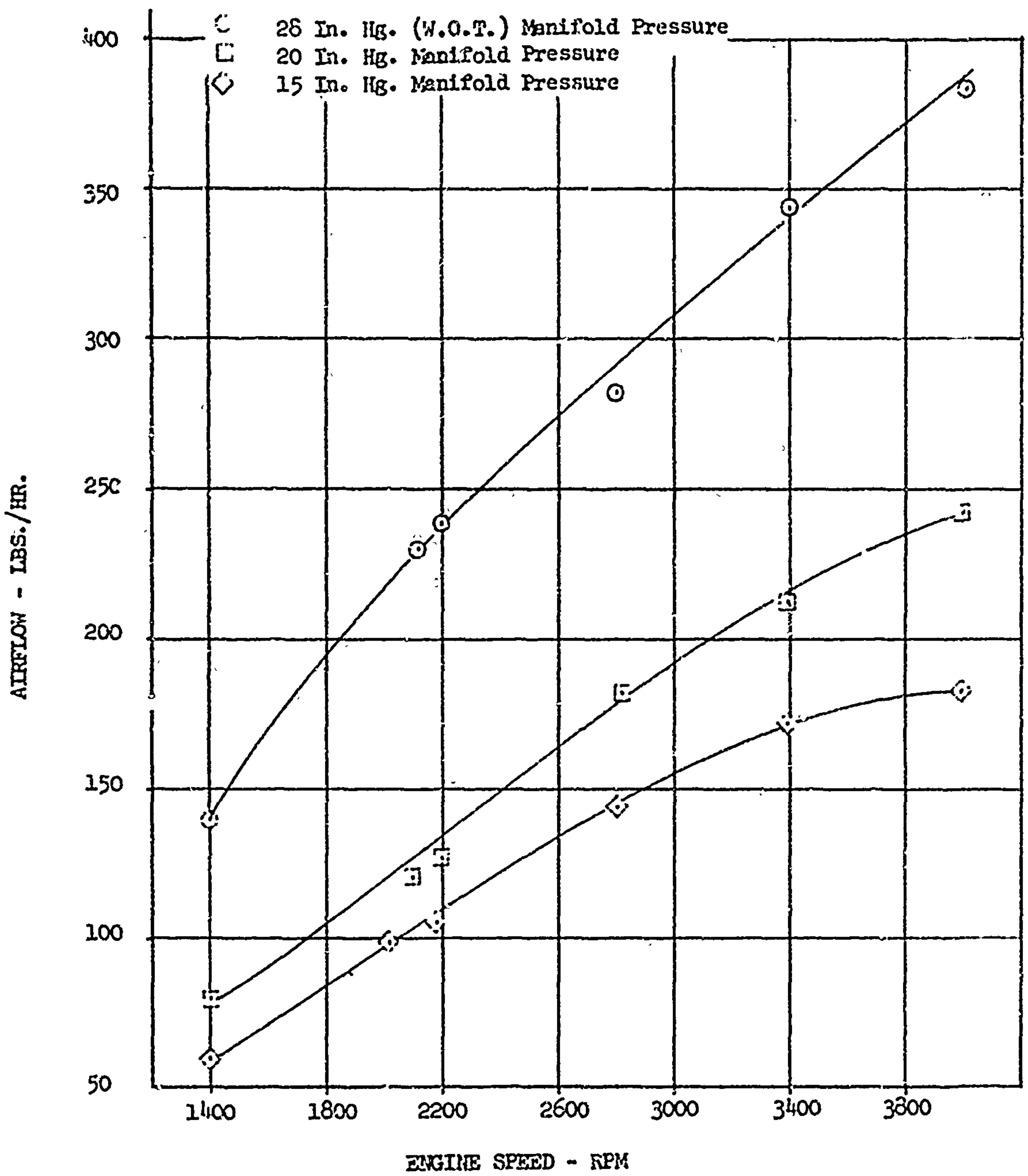





Fig. 26

Comparison of Optimum Output with the Mixing Chamber to the Zenith $NH_3$ Carburator at 11.7:1 C.R.

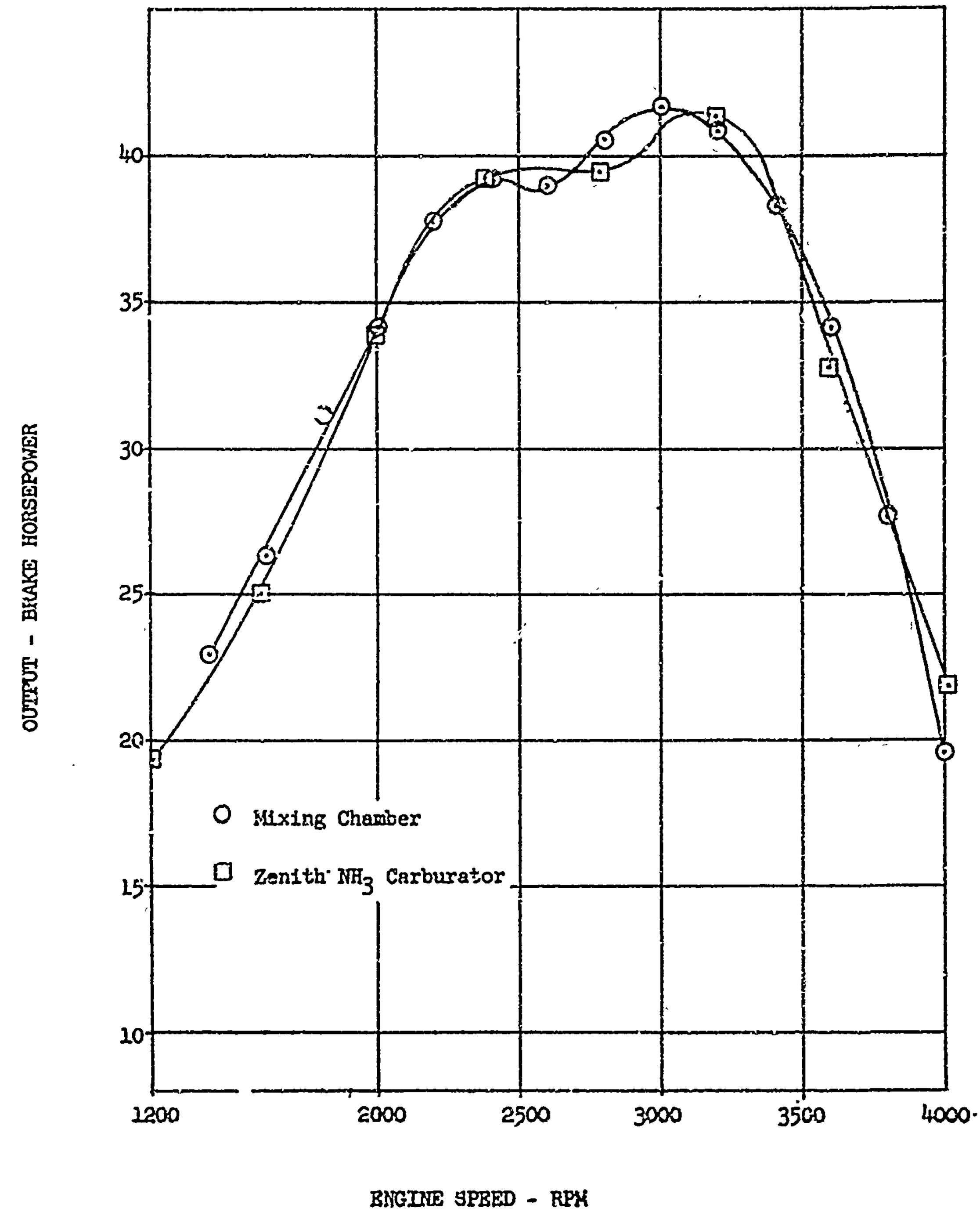


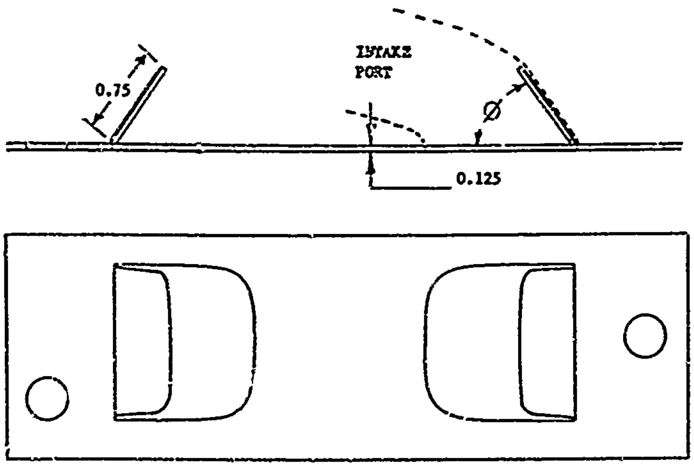

Fig. 27

DISCUSSION

mixture. Swirl deflectors, which could be installed in the induction air ports, see Fig. 28, were fabricated and installed. With the angle of the deflectors optimized, power obtainable was increased throughout the speed range when operating with ammonia vapor only. More important, the maximum speed attainable was increased by 200 rpm. With hydrogen added to the ammonia vapor, there was no increase in available power; but the hydrogen quantity required for satisfactory performance was reduced by 25 percent or more throughout the speed range. Figure 29 is a graphic presentation of this data.

## Effect of Temperature

An arrangement was made to vary the temperature of the induction air and of the ammonia vapor being supplied to the engine. The latter was considered of prime importance for determining cooler size in designing an ammonia dissociator sub-system. Tests showed that increasing the ammonia vapor temperature to 260°F before the carburetor resulted in a power loss of six per cent compared to power attained with fuel to carburetor of 75°F. Variations in induction manifold temperature (as distinguished from induction air temperature before carburetor) had only slight effect on performance. Optimum manifold temperature was in the range of 135 to 140°F.




NOTE:

1. Swirl deflectors mounted between cylinder head and intake manifold so the deflector protrudes into the intake port

2. Angle Ø for cylinder #1 = 55°
   2 = 50°
   3 = 50°
   4 = 55°

Fig. 28. Sketch of L-141 Swirl Deflectors.

Effects of Swirl on Ammonia Engine Performance

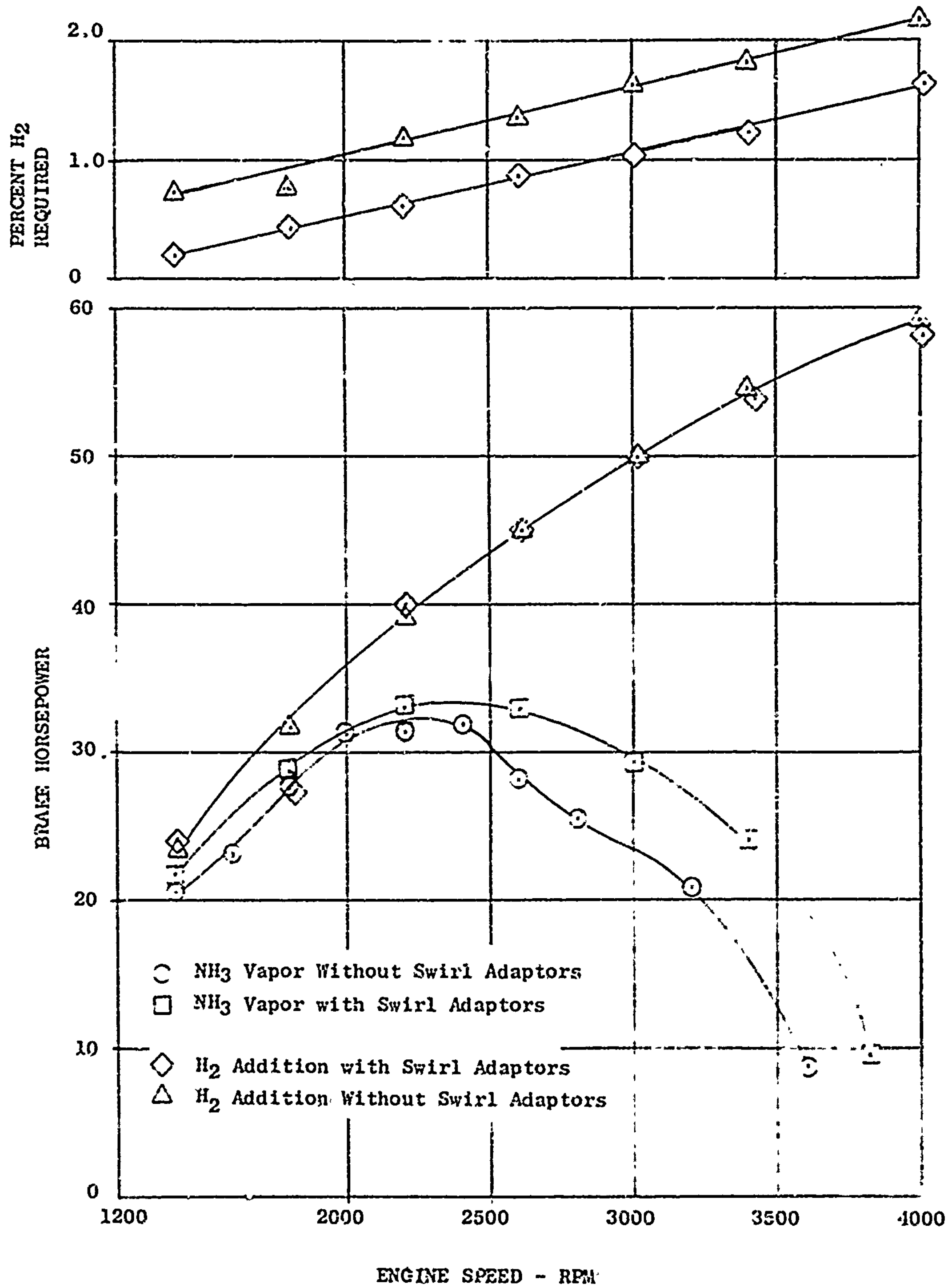





Fig. 29

## DISCUSSION

### Pre-Combustion Chambers

A series of precombustion chambers, varying in size from 3.4 percent to 39.2 percent of the total clearance volume at top dead center, were installed and tested. These chambers were all scavenged and charged via the main combustion chamber. Performance was not satisfactory, partially because of poor scavenging of the pre-chamber, and partially because the low rate of flame propagation of ammonia - air mixture does not develop a sufficient "torch" to ignite the main charge.

Precombustion chambers of the "Walker" type, Reference 5, having a separate supply of carbureted gasoline as an auxiliary fuel in addition to the ammonia - air mixture from the main combustion chamber, were manufactured and installed for testing. These, likewise, were not satisfactory in the sizes and designs evaluated. This does not preclude the possibility that a pre-chamber could be designed for satisfactory operation in an ammonia engine.

### Additives

Many additives to aid combustion and rate of flame propagation of ammonia - air mixtures have been investigated by Continental and others (Reference 4) with the result that hydrogen and acetylene are considered the most effective. The quantities of acetylene, however, were considered to be excessive (22 percent by weight); in addition, hydrogen is available from the same fuel source as the ammonia, or can be generated on the engine by using a thermal dissociator. For these reasons it was decided that engine test work should be restricted to use of hydrogen as a fuel additive.

Verkamp et al, (Reference 4, have shown that, in addition to reducing the ignition energy required to fire ammonia, small amounts of hydrogen greatly broaden the fuel-air ratio where combustion can be sustained. Samuelsen, Reference 2, has shown that small amounts of hydrogen can substantially improve the flame propagation rate.

Hydrogen addition did, in effect, cause dramatic improvements in engine performance, particularly at high speeds. Other changes to the engine to improve combustion, however, had just as dramatic an effect on the requirements for hydrogen addition. For example as the engine was set-up early in the program with 12.6:1 compression ratio and the standard L-141 ignition system, the hydrogen requirement at full throttle was 2.2 percent by weight at 1200 rpm and 5.0 percent at 4000 rpm.

## DISUSSION

Use of a Mallory U12A ignition coil reduced the need for hydrogen under wide open throttle conditions to 1.2 percent at 1200 rpm, and 2.8 percent at 3600 rpm.

Installation of the Mallory "Super Mag" reduced the hydrogen requirement at full throttle to 0.6 percent at 1400 rpm, and 2.2 percent at 4000 rpm.

Introduction of long reach spark plugs and swirl deflectors to the induction air decreased the need for hydrogen still further (none at all) at all speeds up to 1600 rpm, and only 1.6 percent at 4000 rpm.

It is unfortunate that the absolute optimum performance could not be obtained; because, when the best spark plug combination was obtained, the peak cylinder pressures were excessive even without hydrogen addition and it was necessary to reduce the compression ratio to 10.26:1. However, with the best ignition components and the lower compression ratio, the hydrogen requirement at full throttle had still been reduced to 1.5 percent at 4000 rpm.

At one time provisions were made for introducing a small amount of gasoline into the induction air prior to the ammonia carburetor. With this arrangement it was possible to equal the power output when operating with hydrogen in the ammonia - air mixture. The maximum amount of gasoline (eight pounds per hour) was considered excessive so no further effort was expended in this direction.

Partial recirculation of the exhaust gases to improve combustion was also attempted with no noticeable effect.

### Dissociator Sub-System

As a means of generating hydrogen right at the engine, a thermal - catalytic dissociator subsystem was proposed. Allison Division of General Motors Corporation was commissioned to design, fabricate, and develop such an arrangement for the L-141 engine. The subsystem consists of a dissociation products cooler which will partially vaporize the incoming ammonia and, at the same time, cool the dissociation products from the dissociator. A preheater - dissociator, which is heated by the engine exhaust, heats the ammonia to dissociation temperature and decomposes part of the ammonia into nitrogen and hydrogen. Flow diagrams and details of design of the dissociator subsystem are covered in detail in Appendix III, Volume II.

## DISCUSSION

The first unit delivered to Continental used a sintered triply-promoted iron catalyst in the dissociator section. When the dissociator - subsystem was installed on the L-141 engine for testing, a substantial improvement in performance was observed. Two major problems developed, however:

1. The engine was not flexible nor responsive to changes in load or speed.

2. The catalyst broke down into fine powder which blocked the tubes of the vaporizer.

A further explanation of problem (1.) is in order. The system becomes effective and starts producing hydrogen, after about 15 minutes of operation. As the dissociator began to function and produce hydrogen, the exhaust temperature increased and the dissociator became more effective; the action was cumulative. However, the complete system was very sensitive to sudden increases in ammonia flow because the vaporization of ammonia apparently cools the complete system excessively, which reduces the hydrogen output and exhaust temperature, and operation of the dissociator collapses; again the process is cumulative.

The dissociator subsystem was completely redesigned using porous nickel (Foametal) as the catalyst. This second generation subystem included an electrically - heated auxilliary dissociator for use in starting and under light load conditions. The second generation dissociator subsystem was installed on the L-141 engine and subjected to preliminary testing. The results of these tests are shown in Fig. 30.

Performance in general was not satisfactory. The flowpaths of the dissociator are rather complex, see Fig. 31, with several valves to regulate the flow pattern and, in turn, control the temperature of the ammonia vapor to the carburetor and the percentage of free hydrogen generated. With the various valves set for proper operation at full load and speed, the system would not warm up at no load conditions, even with the electric dissociator energized. The dissociator would not warm up at simulated road load conditions up to 3000 rpm (approximately 55 mph).

At wide open throttle (WOT) and with the engine lugging, warm up to produce 3.0 percent hydrogen could be achieved in about 10 minutes. Under these conditions, with the dissociator subsystem well warmed up, the engine

| Engine Speed (RPM) | Manifold Pressure ("Hg.) | Exhaust Temperature (°F) | Power to Auxiliary Dissociator (watts) | Fuel to Carburetor Temperature (°F) | Percent Hydrogen By Weight |
|---|---|---|---|---|---|
| 1200 | 29.1 | 1040 | 0 | 84 | 0.4 |
| 1200 | 29.1 | 980 | 670 | 91 | 2.1 |
| 1600 | 29.0 | 1160 | 0 | 88 | 1.45 |
| 1600 | 15.0 | 960 | 0 | 94 | 0.25 |
| 1600 | 28.8 | 1085 | 680 | 92 | 2.3 |
| 1600 | 15.1 | 945 | 760 | 90 | 2.75 |
| 2000 | 28.8 | 1200 | 0 | 94 | 2.1 |
| 2000 | 28.6 | 1200 | 720 | 93 | 2.5 |
| 2000 | 15.0 | 1000 | 650 | 90 | 2.0 |
| 2400 | 28.7 | 1265 | 0 | 90 | 2.6 |
| 2400 | 28.4 | 1240 | 730 | 88 | 2.7 |
| 2800 | 28.4 | 1310 | 0 | 90 | 3.0 |
| 2800 | 28.2 | 1355 | 630 | 76 | 2.7 |
| 3200 | 28.2 | 1400 | 0 | 77 | 3.2 |
| 3200 | 27.9 | 1430 | 500 | 68 | 2.5 |
| 3600 | 29.8 | 1400 | 0 | 76 | 3.2 |
| 4000 | 27.8 | 1430 | 0 | 74 | 3.2 |
| 4000 | 27.7 | 1430 | 490 | 54 | 2.3 |

Fig. 30. Engine Test Data With Dissociator.

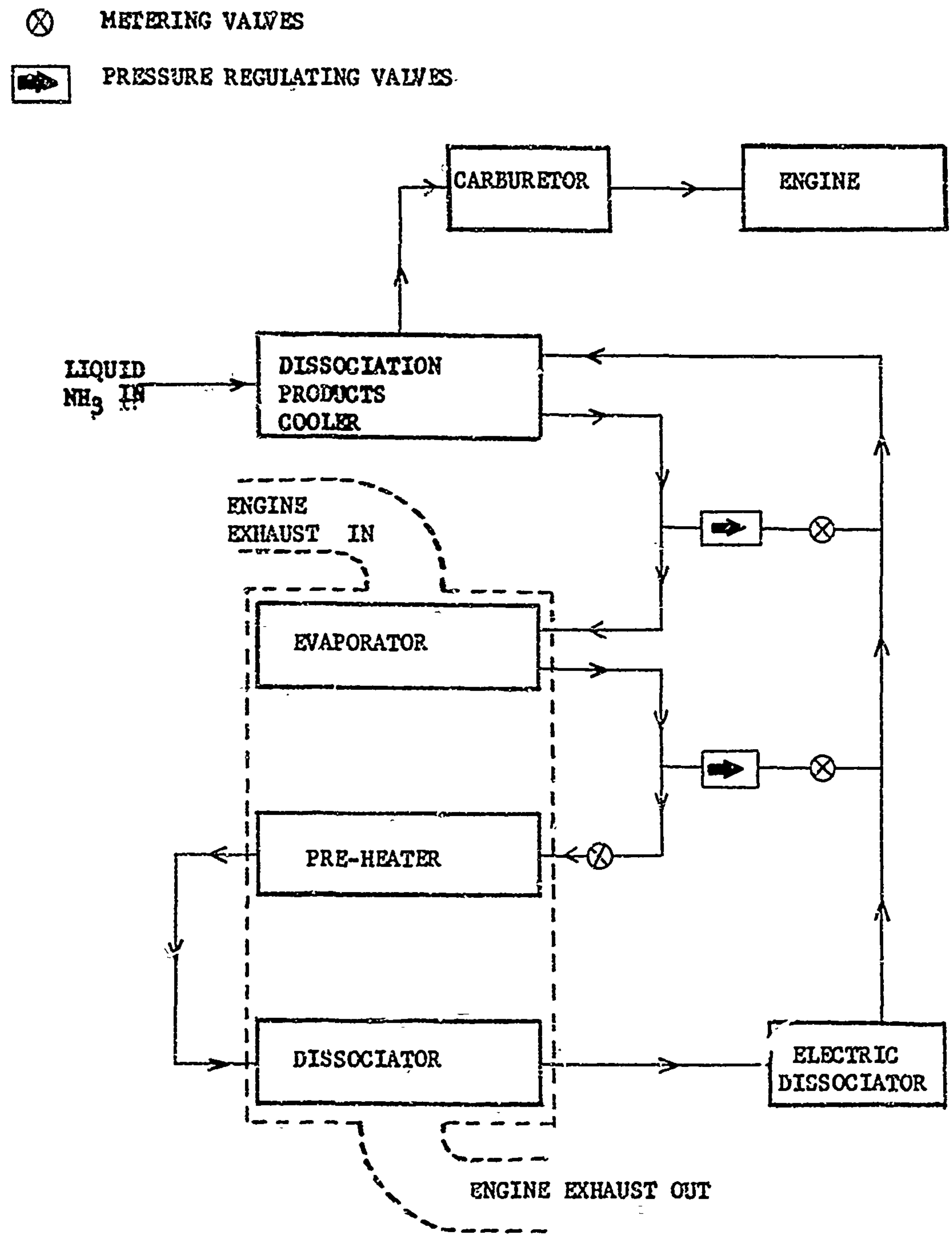



Fig. 31. Schematic Arrangement of Dissociator Subsystem.

DISCUSSION

could be throttled back to low load and speed and would then respond readily and accelerate rapidly under load. If, however, the engine was allowed to remain at low load and speed for any appreciable time, the dissociator subsystem would cool down and acceleration would then be very sluggish.

In view of the fact that the hydrogen requirements for this engine have been reduced from five to 1.5 percent, the dissociator subsystem as designed and fabricated under this investigation is of excessive size and mass. The engine, as now conceived, can start, run, accelerate, pull over 40 horsepower at 3000 rpm and deliver over 20 horsepower at 4000 rpm WITHOUT THE USE OF ANY HYDROGEN.

In light of the above facts, it is considered that a redesigned dissociator subsystem, which would be primarily a fuel vaporizor and need to generate only small amounts of hydrogen to top off the high speed, high load area, could be highly successful and improve engine flexibility.

SUMMARY

The L-141 engine, as finally developed to burn anhydrous ammonia, consisted of the standard engine with the following modifications:

1. Mallory Super-Mag Magneto in lieu of Auto-Lite igniter.

2. Champion OJ-22-1 Spark Plugs.

3. Swirl deflectors in induction air ports.

4. Zenith Model PC1-10 ammonia vapor carburetor.

5. Special Cylinder Head, CAE part No. 594015, modified to give 10.26:1 compression ratio.

While performance could be improved by using a higher compression ratio, it was not possible to utilize the higher ratio because peak cylinder pressures would be so high as to cause crankshaft failure by bending. The engine was assembled as indicated above and operated to determine final performance characteristics.

## DISCUSSION

Figure 32 is a fuel map showing the maximum horsepower versus rpm that could be obtained with ammonia vapor alone (no hydrogen addition). Over-laid on this curve are lines of constant brake specific fuel consumption.

Figure 33 is a plot of horsepower output versus rpm for constant manifold pressures, for the most part with ammonia vapor alone. In addition lines of constant manifold pressure have been added to show performance with hydrogen to top off the high-load, high-speed area. Hydrogen requirements at full throttle varied from 0.6 percent at 2200 rpm to 1.54 percent at 4000 rpm. Slightly higher amounts, up to 2.1 percent, were required at some part throttle conditions.

## COMPRESSION-IGNITION INVESTIGATION

### Direct Injection of Ammonia

The AVDS-1790 Vee Twin was rebuilt using 30:1 compression ratio pistons. This was the highest compression ratio possible without changing timing of the valve events. The piston had a spherical dome to match the contour of the cylinder head, see Figs. 34 and 35, with cylindrical cut-outs for the valves and the fuel injection nozzle tip. Two single cylinder American Bosch Model APE-1BB-1300-X5751A fuel injection pumps with 13 mm plungers were installed to ensure adequate capacity for pumping liquid ammonia and to permit better control of each cylinder. The engine as assembled was checked out by both motoring and running at light load on diesel fuel.

Considerable difficulty was encountered in the first attempts to run the engine using liquid anhydrous ammonia as fuel because the fuel would vaporize before getting to the fuel injection nozzles. Several changes were made to the fuel system to improve its pumping characteristics, including the following:

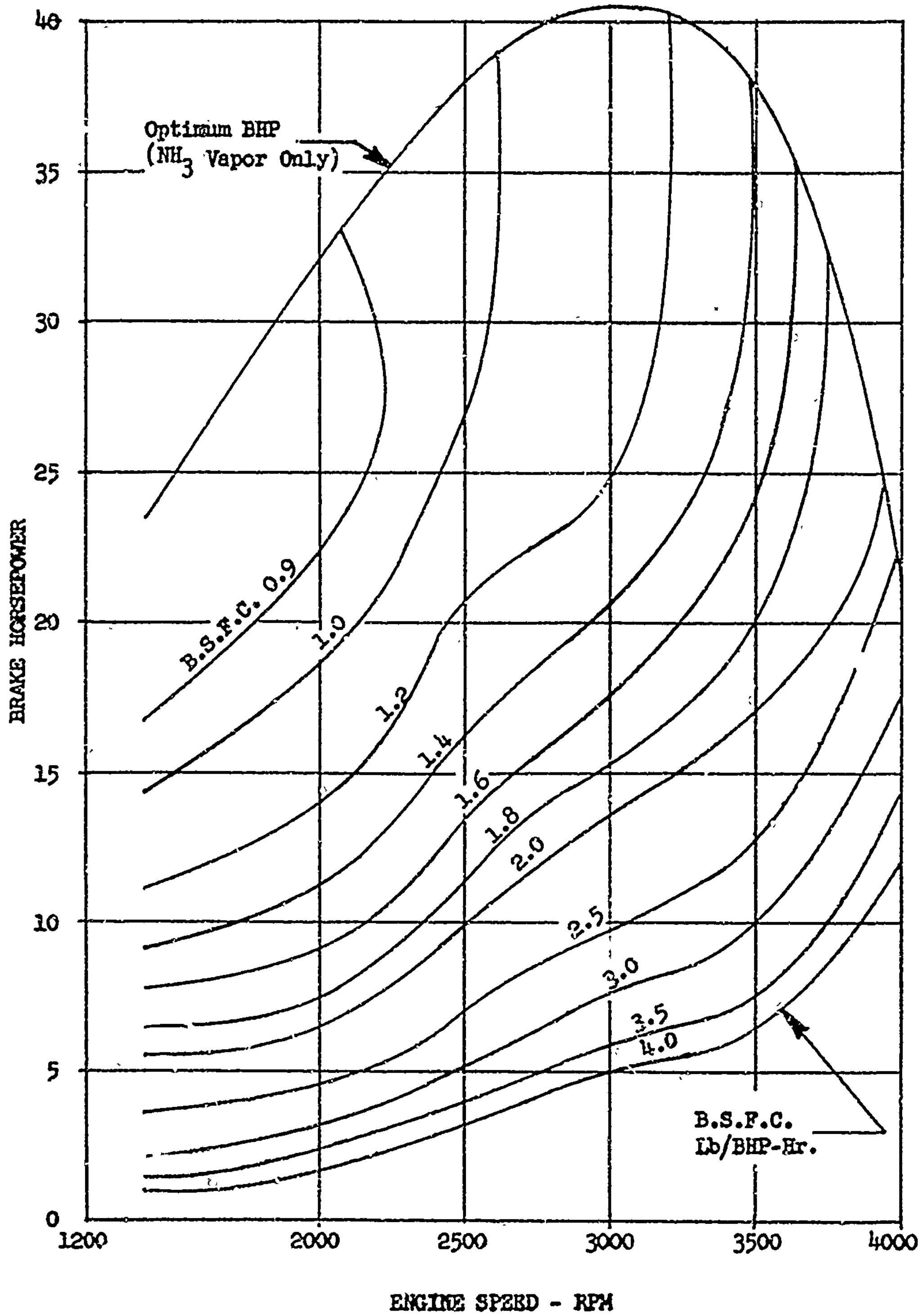
1. Reduced nozzle valve opening pressure from 3000 to 2200 psi. No improvement.

2. Changed nozzle drain lines to vent to atmosphere in lieu of returning to intake side of fuel injection pump. This permitted injection of some ammonia at speeds up to 1000 rpm and with full rack setting on the pump. The fuel injection pump became vapor bound at higher speeds or at partial rack settings.

L-141

Performance with Anhydrous Ammonia Fuel Using a Mallory Magneto, Champion OJ-22-1 Plugs (Full Reach), Swirl Adaptors, $NH_3$ Carb. and 10.26:1 Compression Ratio

J. Pinter



$NH_3$-236

Ammonia Engine Performance In Final Configuration

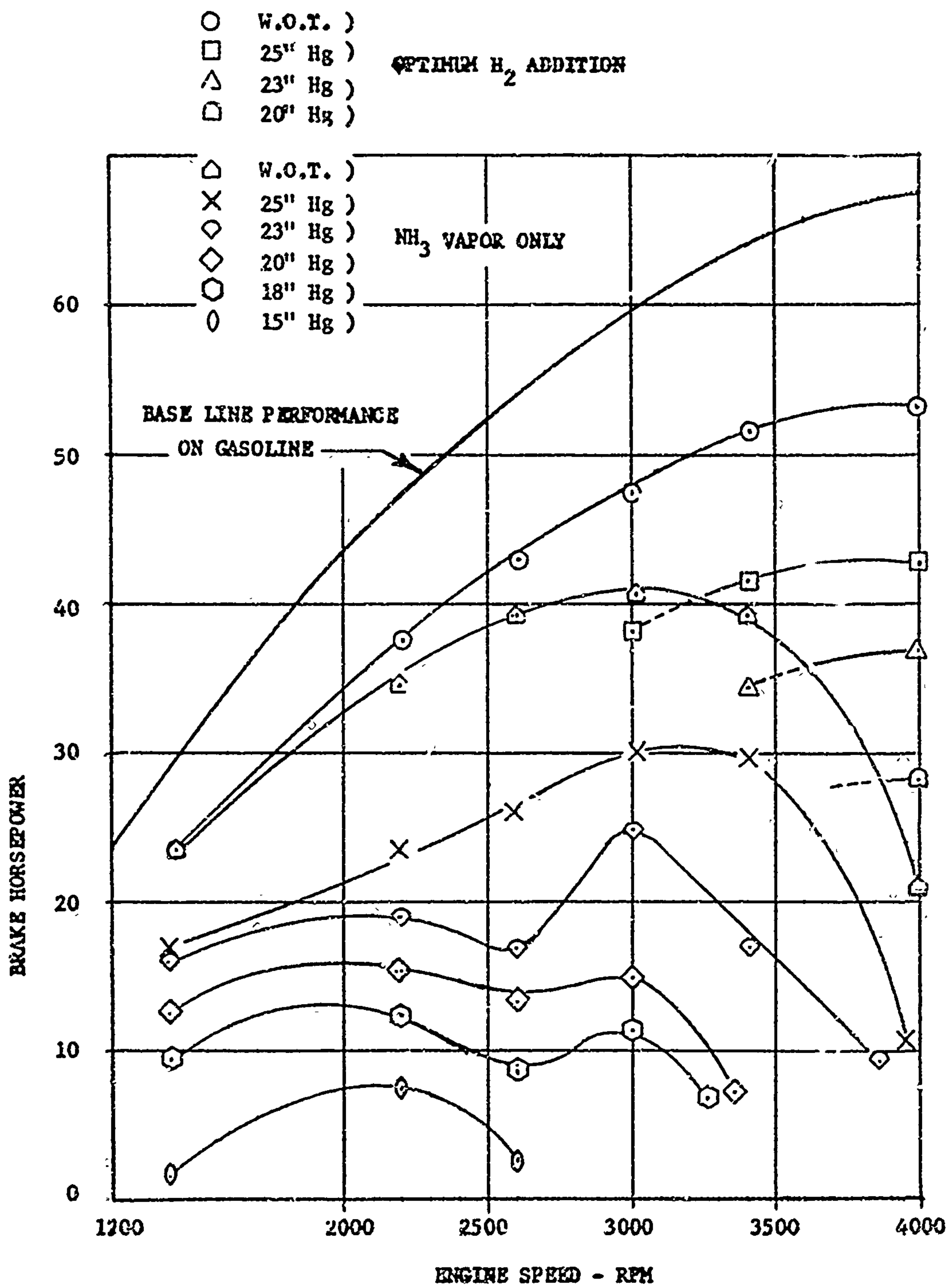



Fig. 33

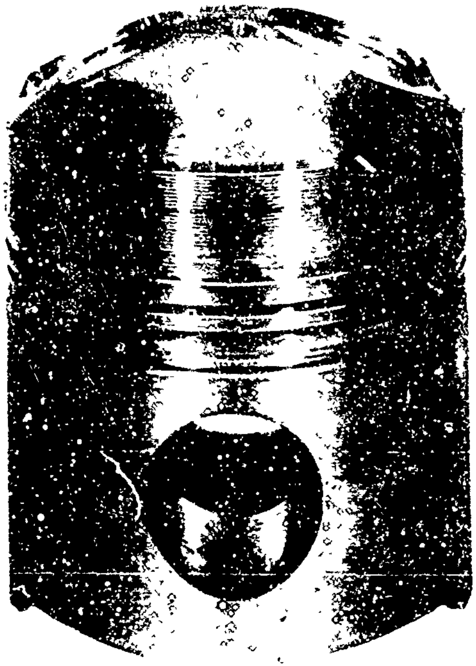


Fig. 34. Side View of 30:1 Compression Ratio Piston. (D-35192)

Fig. 35. Top View of 30:1 Compression Ratio Piston. (D-35193)

DISCUSSION

3. Packed fuel injection pump in dry ice. This permitted pumping of ammonia under all speed conditions; however, there was not sufficient refrigerating capacity for prolonged operation.

4. A heat exchanger was fabricated using 10 feet of one-inch stainless steel tubing in a sheet metal box filled with alcohol and dry ice. This device cooled the liquid ammonia, before the fuel injection pump, to a temperature of 0° to 20°F, depending upon fuel flow. With this arrangement, it was possible to pump and inject liquid ammonia consistently.

<u>Performance</u>

While going through the above procedures to assure pumping of liquid ammonia, there was some question as to whether the lack of combustion was due to inability to pump ammonia, or lack of ignition from compression temperatures. To settle this point, the engine was operated with ammonia vapor aspirated into the induction air system. Compression-ignition was achieved at low speeds, but the engine was not self-sustaining above 1200 rpm. With gaseous hydrogen added to the ammonia vapor, the engine was self-sustaining up to 2300 rpm. No high level of power output could be maintained for two reasons:

1. Compression pressures at 30:1 compression ratio, naturally aspirated, were 1200 psi which allowed only about half as much range for pressure increase due to combustion.

2. Compression-ignition was uncontrolled from a timing point of view and resulting combustion was extremely rough.

When the engine was operated at room temperature and liquid anhydrous ammonia was injected into the combustion chamber, no combustion at all was observed. The engine was then motored at high speed with heated induction air so as to heat the engine until normal operating temperatures were simulated. Typical temperatures and pressures are as shown in Table I. The injection pump racks were then opened so as to inject small quantities of ammonia.

## DISCUSSION

### TABLE I

### TYPICAL OPERATING CONDITIONS PRIOR TO AMMONIA INJECTION

| rpm | 1000 | 1200 | 1700 | 1800 | 2000 |
|---|---|---|---|---|---|
| Manifold Pressure, "$H_g$A | 31 | 34 | 31 | 34 | 33 |
| Manifold Temperature, °F | 197 | 190 | 214 | 208 | 208 |
| Cylinder Heat Temperature, °F | 326 | 343 | 374 | 379 | 408 |
| Oil Temperature, °F | 182 | 182 | 186 | 186 | 193 |
| Compression Pressure, psi | 1150 | 1250 | 1300 | 1350 | 1300 |
| Exhaust Temperature, °F | -- | 180 | 250 | 228 | 280 |

On the first injection, a slight increase in cylinder pressure could be observed, indicating that part of the fuel charge was trying to burn. Subsequent injections of ammonia, however, refrigerated the combustion air to the point where compression pressures were lowered as much as 250 psi. Analysis of the indicator cards indicates that compression temperatures with ammonia injection were in the neighborhood of 1000°F, about 200 degrees below the spontaneous ignition temperature of ammonia. At this point, the direct injection approach appeared to be so impractical that it was terminated, with the advice and consent of the ATAC Project Engineer.

## COMPRESSION-IGNITION INVESTIGATION

### Diesel Pilot Injection

An AVDS-1790 Vee-Twin (a two-cylinder version of one bay of the 12-cylinder tank engine) was installed in Continental Test Cell No. A-7. The engine was equipped with the latest design pistons having a compression ratio of 18.6:1. The production 12-cylinder fuel injection pump was used with 10 of the fuel injectors injecting into dummy tanks, from which the fuel could be returned

DISCUSSION

to the supply system. The engine was given a green-run on number one diesel fuel, DF-1, and then was operated to obtain baseline fuel consumption and airflow data at part-load and various speeds. Figures 36 through 41 are part-load performance at 1200 to 2400 rpm. Figure 42 shows full load performance through the speed range.

This engine, being a two-cylinder 90° Vee, has a firing sequence of 270° to 450°. In order to minimize the pulsations in the induction air measuring devices, a large plenum chamber is installed in the induction air lines immediately before the cylinders. This plenum chamber was adapted for use as a mixing chamber for the ammonia and combustion air by simply introducing ammonia vapor into the induction air supply. Ignition was obtained by injecting a pilot charge of diesel fuel into the ammonia-air mixture in the cylinder, similar to the procedure in dual-fuel, natural gas engines.

Diesel engines normally run with considerable excess air because of smoke limitation problems; when running on ammonia, however, the engine wants to operate at air-fuel ratios close to stoichiometric. Due to better air utilization, it was possible to run the engine on ammonia fuel at outputs exceeding diesel fuel output by 32 percent. However, since it was considered that the drive train would limit the amount of torque that could be absorbed, all additional testing was limitedlto the diesel rating.

Figure 43 is typical of the comparison of performance on diesel fuel and on ammonia with pilot injection, while operating within the same limits of peak cylinder pressure and exhaust gas temperature. It is particularly noted that the maximum thermal efficiency with ammonia fuel is 13 percent better than with diesel fuel (51 versus 45 percent), substantiating the predictions of Newhall, Reference 6.

At times, there seemed to be a certain lack in repeatability of test data; this problem was resolved when it was noted the large effect that cylinder head temperature had upon both specific fuel consumption and output as shown in Figure 44. The data for this curve was obtained by running the engine at constant speed, holding the fuel flow and airflow constant and varying the cylinder head temperatures only. There is no ready explanation for this phenomenon other than the possibility of increased dissociation of the ammonia vapor during the compression stroke when running with a hotter cylinder head.

NE$_3$-154

AVDS-1790 V-Twin

Part Load Performance - Diesel Fuel Baseline Data - 1200 RPM - 18.6:1 C.R.
8-Hole .276mm 140° Nozzles - .084-Inch Lines - Inj. Adv. 37° BTC

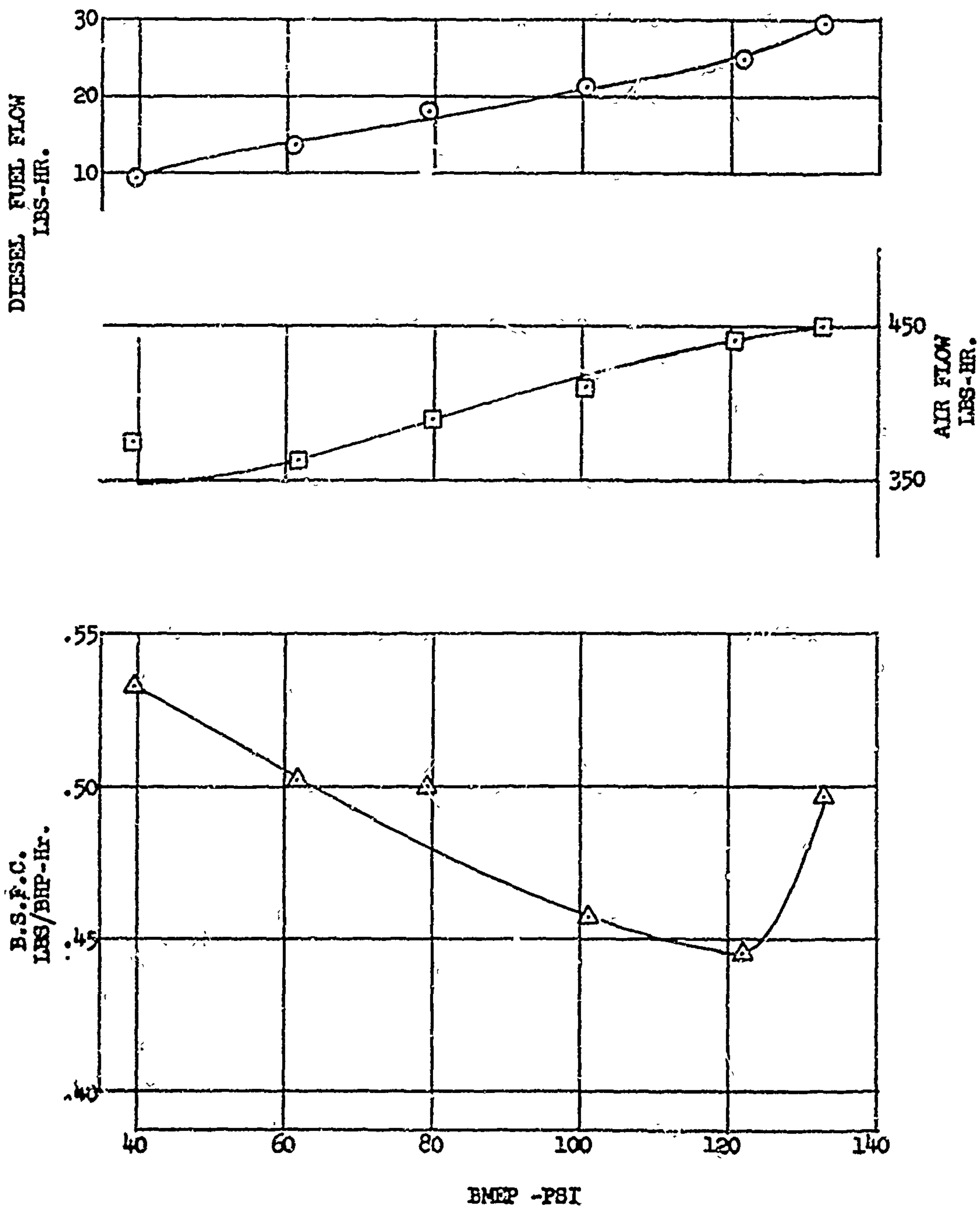





Fig. 36

$NH_3$-155

AVDS-1790 V-Twin

Part Load Performance - Diesel Fuel Baseline Data - 1600 RPM - 18.6:1 C.R.

8-Hole .276mm 140° Nozzles - .084-Inch Lines - Inj. Adv. 41° BTC

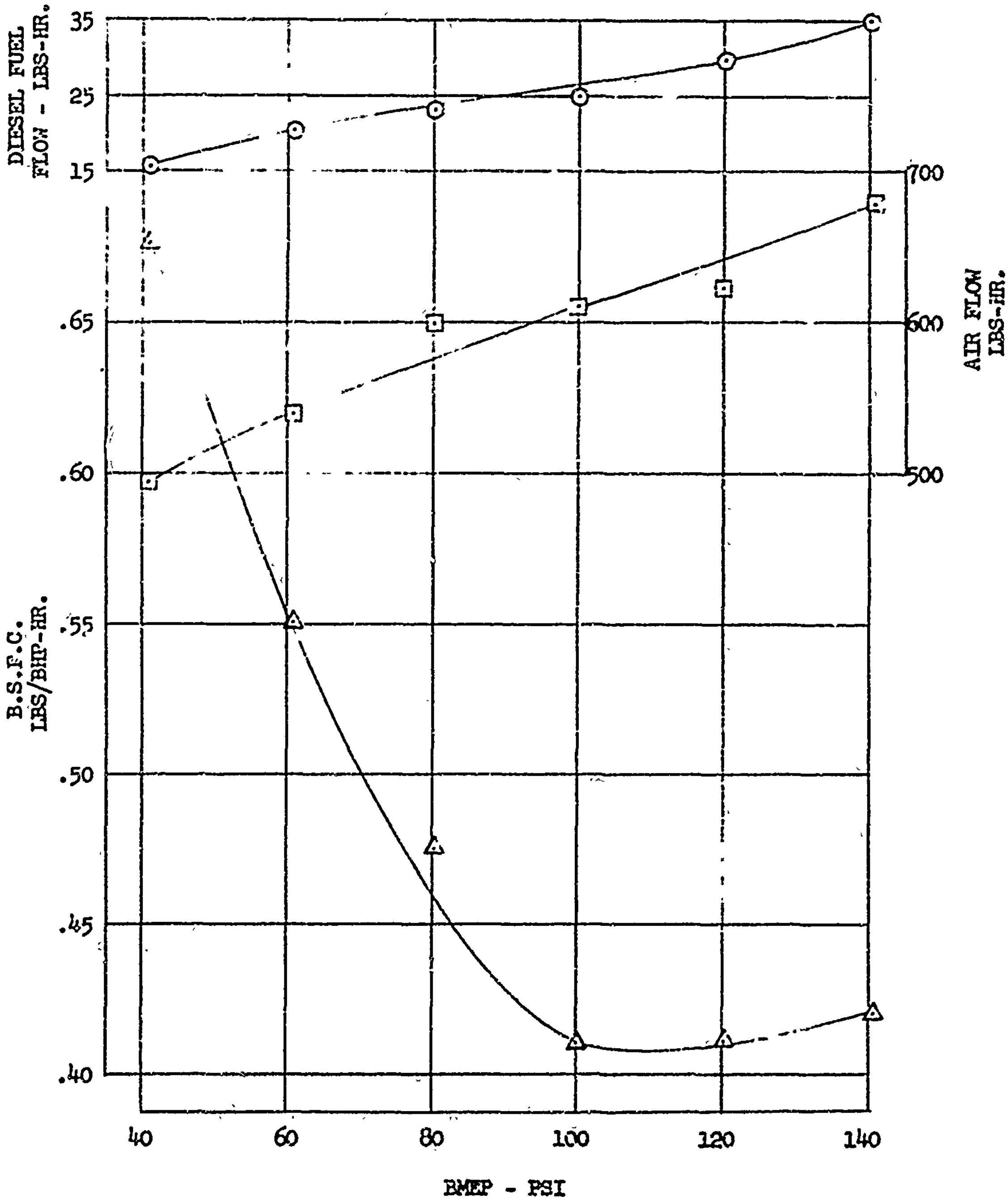





Fig. 37

$NH_3$-156

AVDS-1790 V-Twin

Part Load Performance - Diesel Fuel - Baseline Data - 1800 RPM 18.6:1 C.R.
8-Hole .276mm 140° Nozzles - .084-Inch Lines - Inj. Adv. 43° BTC

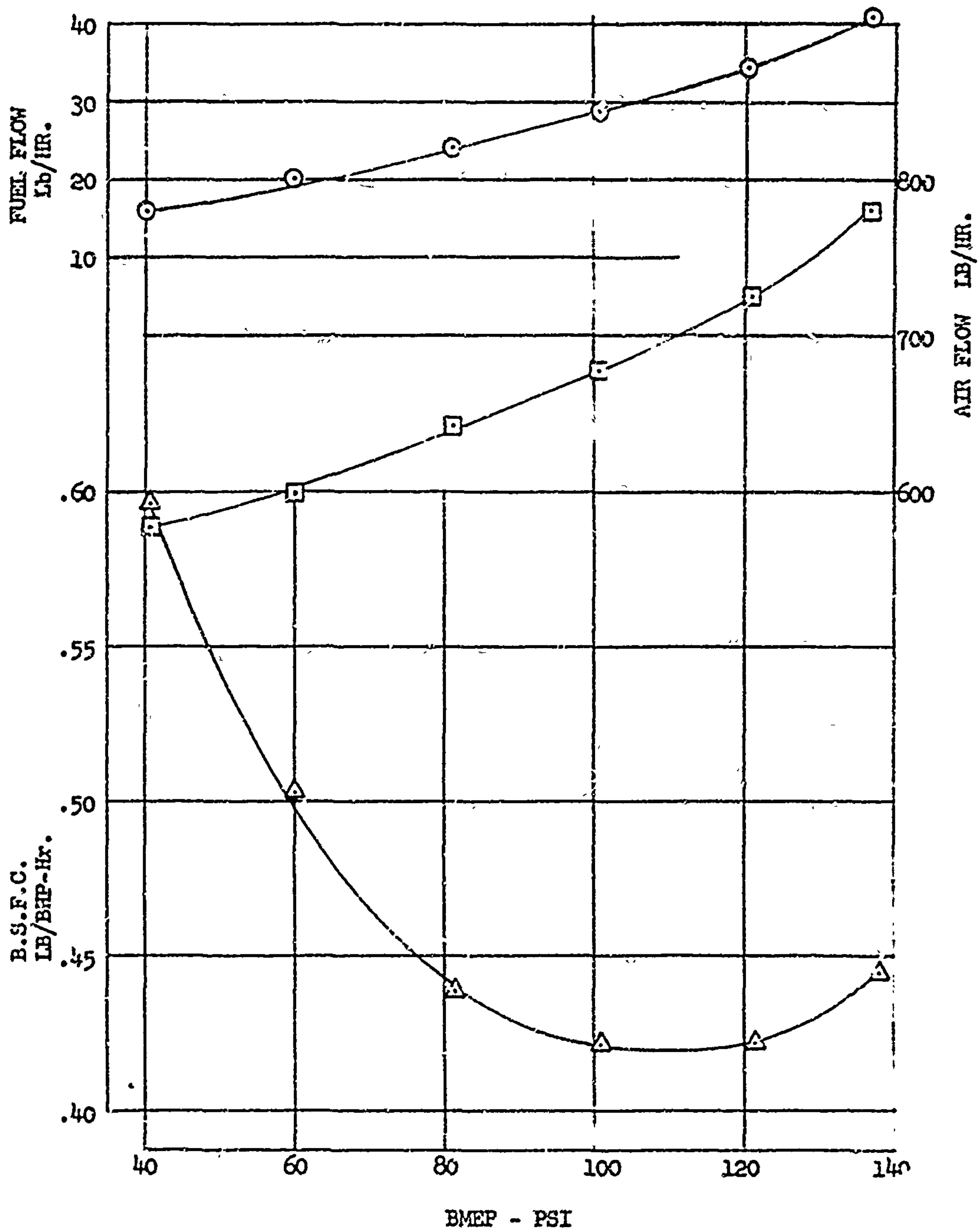



Fig. 38

$NH_3$-157

AVDS-1790 V-Twin

Part Load Performance - Diesel Fuel - Baseline Data - 2000 RPM 18.6:1 C.R.
8-Hole .276mm 140° Nozzles - .084-Inch Lines - Inj. Adv. 43° Crank

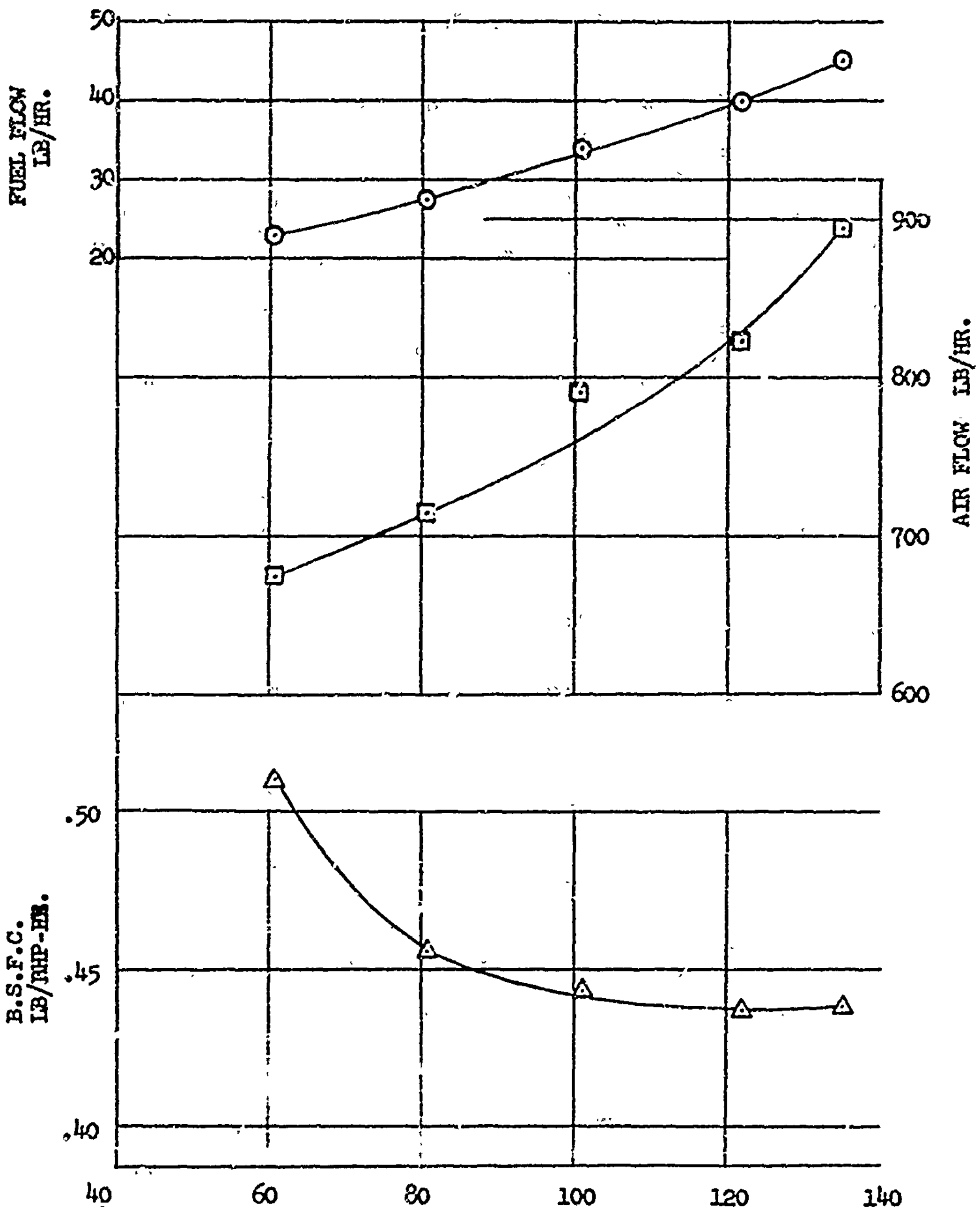





Fig. 39

$NH_3$-158

AVDS-1790 V-Twin
Part Load Performance - Diesel Fuel - Baseline Data - 2200 RPM 18.6:1 C.R.
8-Hole .276mm 140° Nozzles - .084-Inch Lines - Inj. Adv. 43° BTC

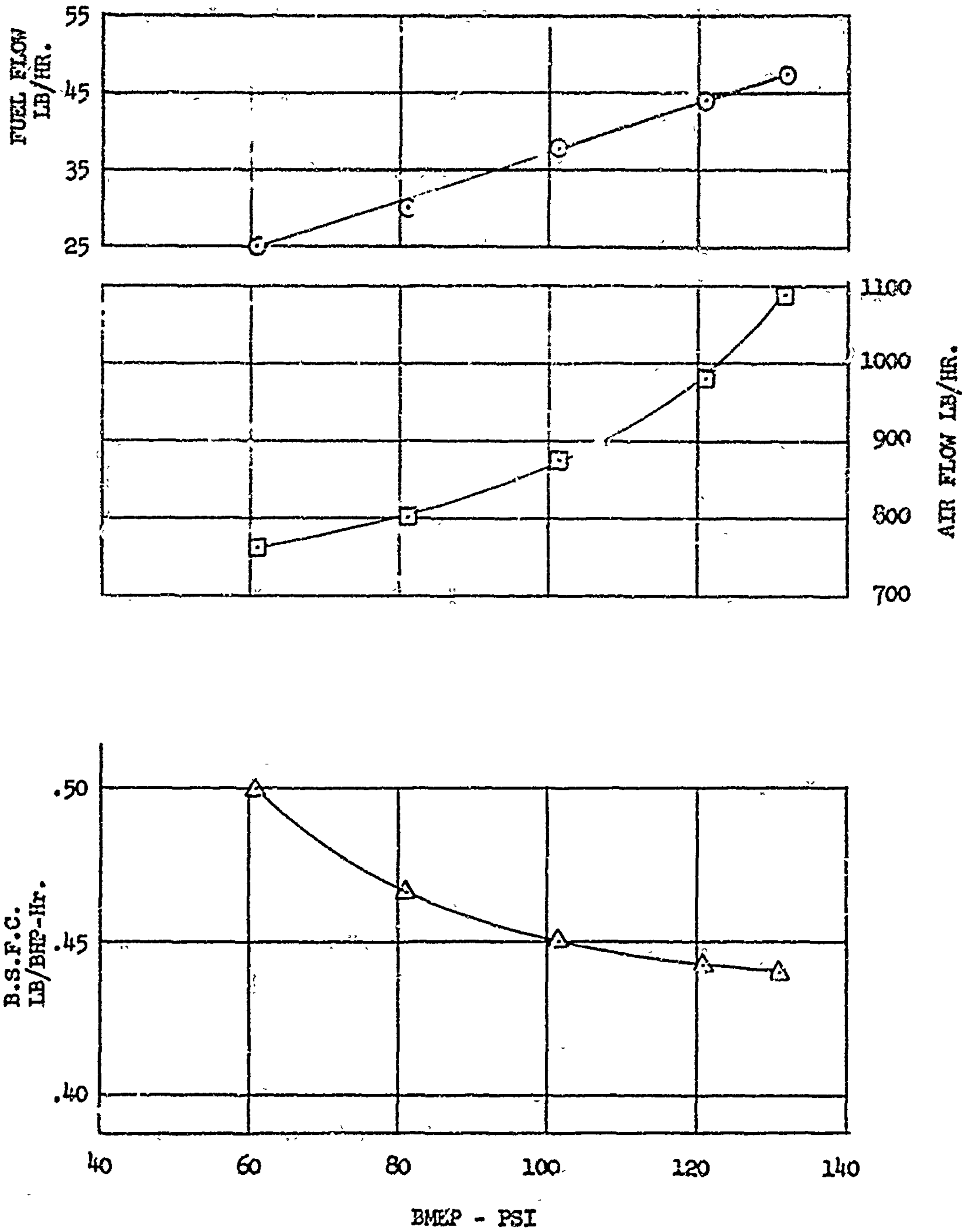



Fig. 40

NH3-159

AVDS-1790 V-Twin

Part Load Performance - Diesel Fuel - Baseline Data - 2400 RPM 18.6:1 C.R.
8-Hole .276mm 140° Nozzles - .034-Inch Lines - Inj. Adv. 43° BTC

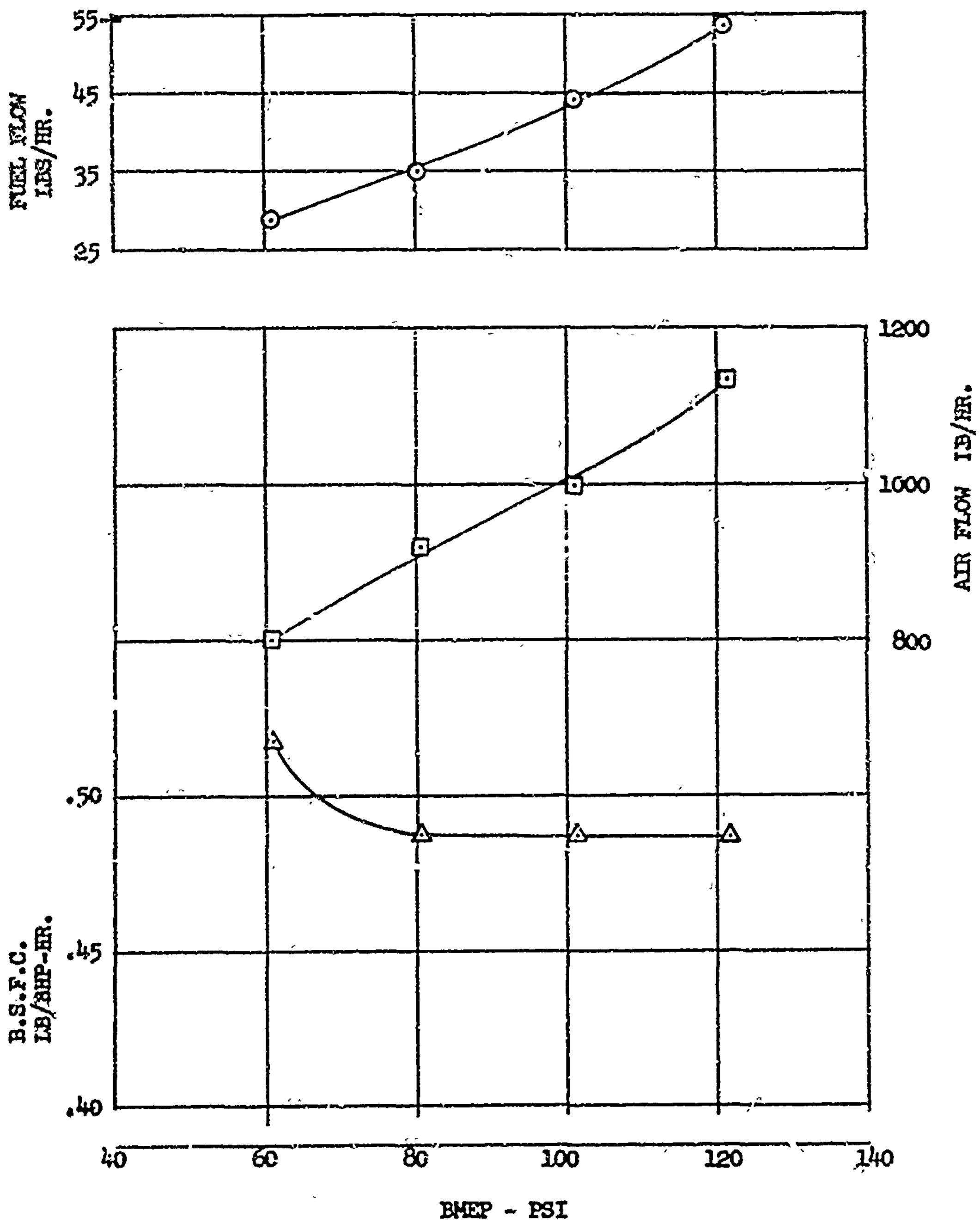





Fig. 41

$NH_3$-169

AVDS-1790 V-Twin
Performance on No. 1 Diesel Fuel - Full Rack 18.6:1 C.R. Factory Air Supercharged - AVDS-1790-2A Injection Pump - 8-.276mm 140° Noz. .084" Lines

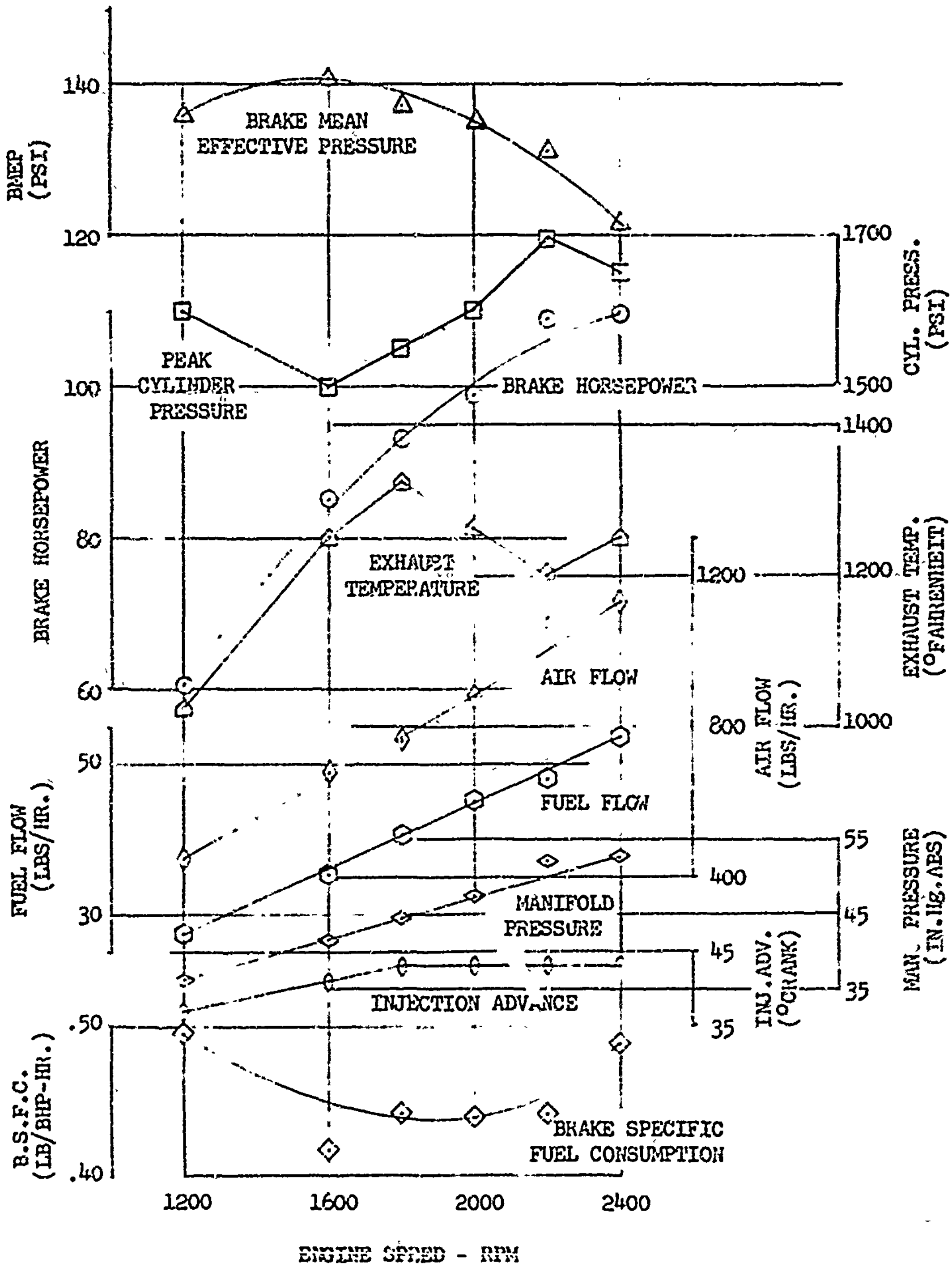





Fig. 42

NH$_3$-139

AVDS-1790 V-Twin (18.6:1 CR)
PRELIMINARY PART LOAD COMPARISON - 1800 RPM

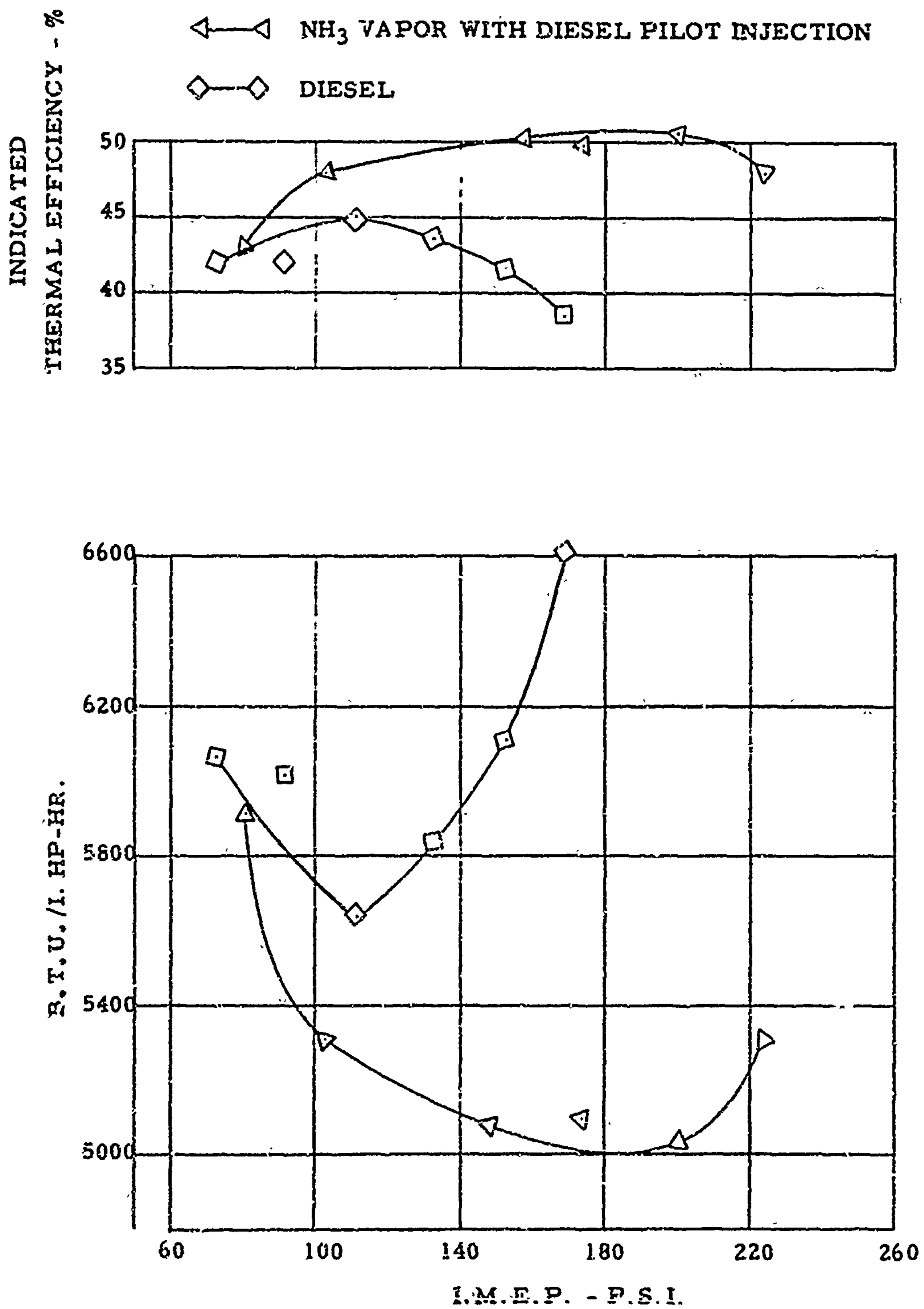





Fig. 43

AVDS-1790 V-Twin

Influence of Cylinder Head Temperature On Specific Fuel Consumption &BMEP

Ammonia Vapor Fuel With Diesel Pilot Injection - 1800 RPM - 18.6:1 CR

D. Hill

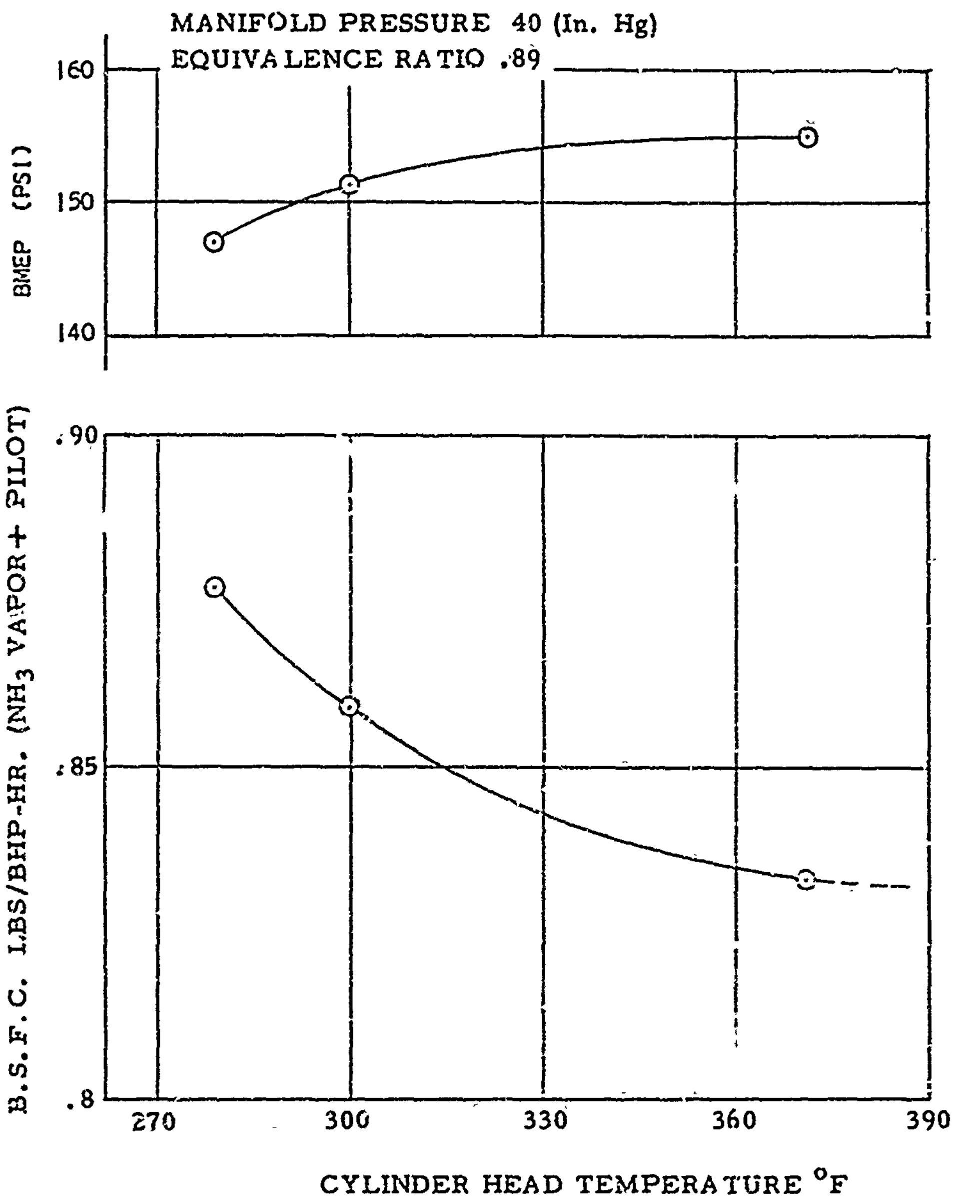





Fig. 44

## DISCUSSION

### Fuel Injection Equipment

Early in this program, it was recognized that for the pilot fuel approach to be considered effective, the amount of diesel pilot fuel used must be held to an absolute minimum. When the fuel rack was backed out to pump small quantities of fuel (less than 10 cubic millimeters per stroke), first one cylinder, then the other, would cease firing. Investigations revealed that the multi-cylinder pumps were not balanced at low fuel flow conditions. American Bosch manufacturers of the fuel injection equipment, advised that their normal acceptable tolerance on these injection pumps was a variation from cylinder-to-cylinder of 7 $mm^3$, much above the minimum it was anticipated could be attained. With this built-in imbalance, it was not possible to lower the minimum pilot fuel at 1800 rpm below 11 $mm^3$ per stroke, even with optimizing the size and number of nozzle holes.

The multi-cylinder fuel injection pump was removed and replaced by two single-cylinder American Bosch, type APE-1BB, pumps with nine millimeter plungers. The pumps were carefully balanced by installing shims so that their plunger lift and port closing characteristics were identical. A series of modifications were made to the fuel injection system to improve the probabilities of pumping minimum quantities of fuel. These modifications included the following:

1. Injection line sizes were changed from 0.084 to 0.055-inch I. D.

2. Zero volume retraction valves were installed in the fuel injection pumps.

3. The four holes in the fuel injection nozzles were successively reduced in diameter from 0.276 mm to 0.22 mm, to 0.20 mm, to 0.19 mm.

   . Figure 45 shows the reduction in pilot fuel requirements versus nozzle hole size. Figure 46 shows the reduction in specific fuel consumption (ammonia only) versus hole size when operating with a relatively constant amount of diesel pilot fuel.

4. Reduced dead fuel volume in nozzle holder.

5. Lapped nozzle needle valves to increase operating clearance.

$NH_3$ -207

AVDS 1790 V-Twin #3
Effect of Nozzle Area on Minimum Pilot Fuel Requirements
With 4 Hole Nozzles at 2400 RPM and 45" AWMP

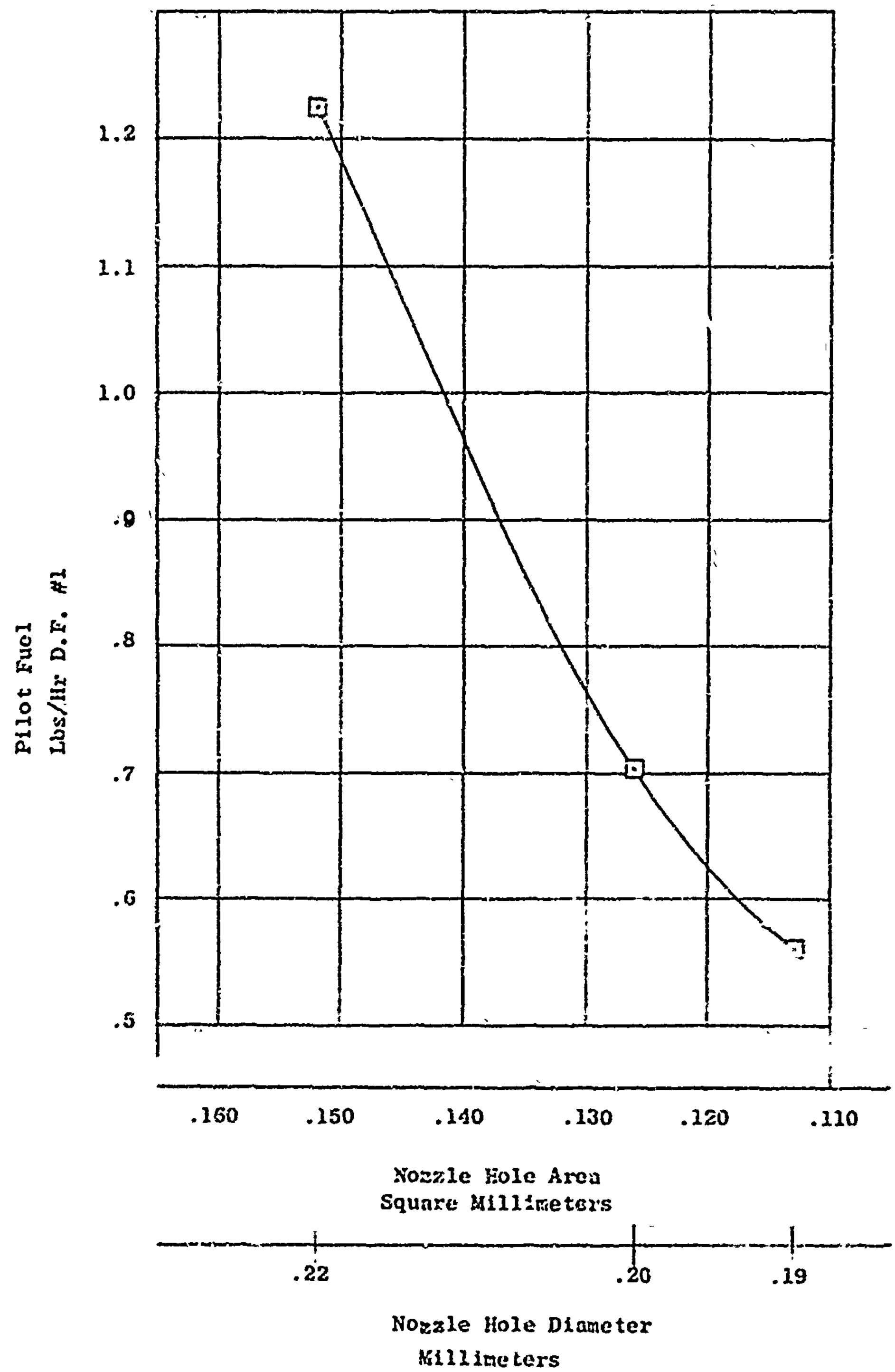





Fig. 45

$NH_3$ - 208

AVDS 1790 V-Twin #3

Effect of Nozzle Hole Area on BSFC with 4 Hole Nozzles at 2400 RPM and 45" ATMP

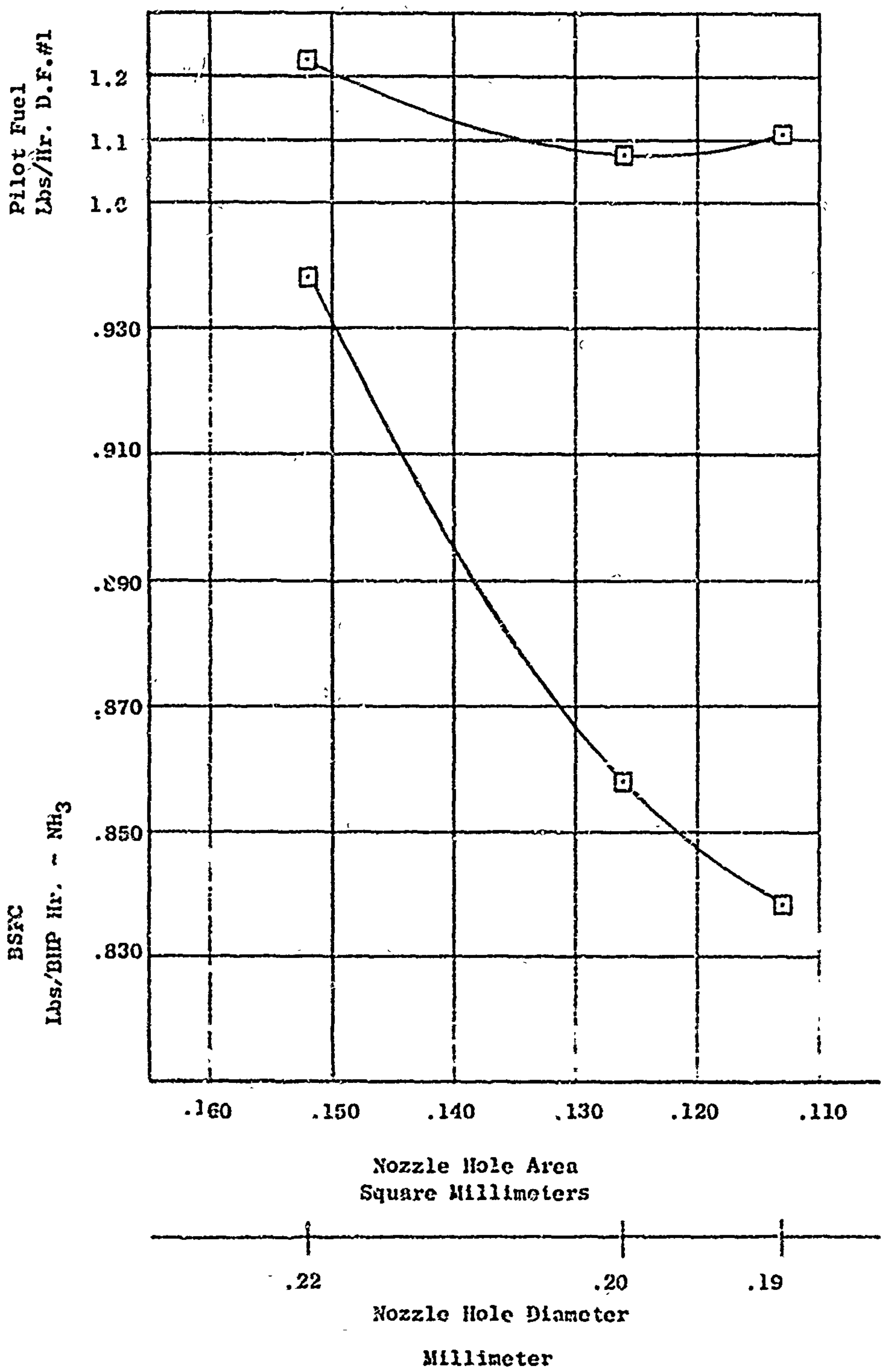





Fig. 46

## DISCUSSION

With all of the above modifications, it was possible to operate the Vee-Twin engine using only 1.8 $mm^3$ per stroke, or 0.35 pounds per hour of diesel pilot fuel at 1800 rpm. Figure 47 is a chronological chart showing the various changes made and the reduction in pilot fuel requirements versus time in weeks.

The reduction in pilot fuel flow was not without drawbacks. The small quantities of fuel being pumped were not sufficient to keep the nozzles cool, resulting in partial plugging of the holes and lacquering of the needle valves. This could occur in as short a period as 15 hours of operation. Figure 48 illustrates the effect of nozzle cleanliness on engine performance. To prevent collecting erroneous data, the fuel injection nozzles were removed and cleaned every morning before beginning the day's operation.

### Manifold Pressure

The Vee-Twin engine was run at 1800 rpm to determine the effect of manifold pressure. Figure 49 shows the effect of manifold pressure on diesel pilot fuel, ammonia flow and load as expressed in indicated mean effective pressure. Optimum ammonia flow is a linear function of manifold pressure, as expected, since the engine wants to run at a fixed stoichiometric ratio. Diesel pilot fuel requirements, however, vary in an inverse fashion, increasing rapidly at manifold pressures below 30 inches of mercury absolute. Between 22 and 17 inches manifold pressure, the engine could carry no useful load, barely generating sufficient power to equal its own parasitic losses. Below 15 inches of mercury manifold pressure the engine was not self-sustaining, due to long ignition lag. Figure 50 is a reproduction of the pressure-time diagrams at 1800 rpm for manifold pressures of 45 to 17 inches of mercury absolute. In all these runs, the optimum fuel injection timing remained the same, but the point of ignition was continually retarded until at 15 inches manifold pressure there was no discernible firing The compression ratio was the same throughout, and theoretically the compression temperatures would have been the same. The increased ignition lag is assumed to be due to the decreased density of the charge and the greater difficulty for a droplet of fuel to find a molecule of oxygen. This situation is similar to the results reported by Boerlage and Broeze in 1932, Referenced Item 7.

### Swirl

Induction air deflectors, capable of increasing or decreasing swirl in the cylinder, were designed and installed. Excessive swirl had a very definite deleterious effect as regards both fuel consumption and power output. Small amounts of swirl appeared to have little, if any effect. Swirl deflectors were no longer used after this preliminary examination.

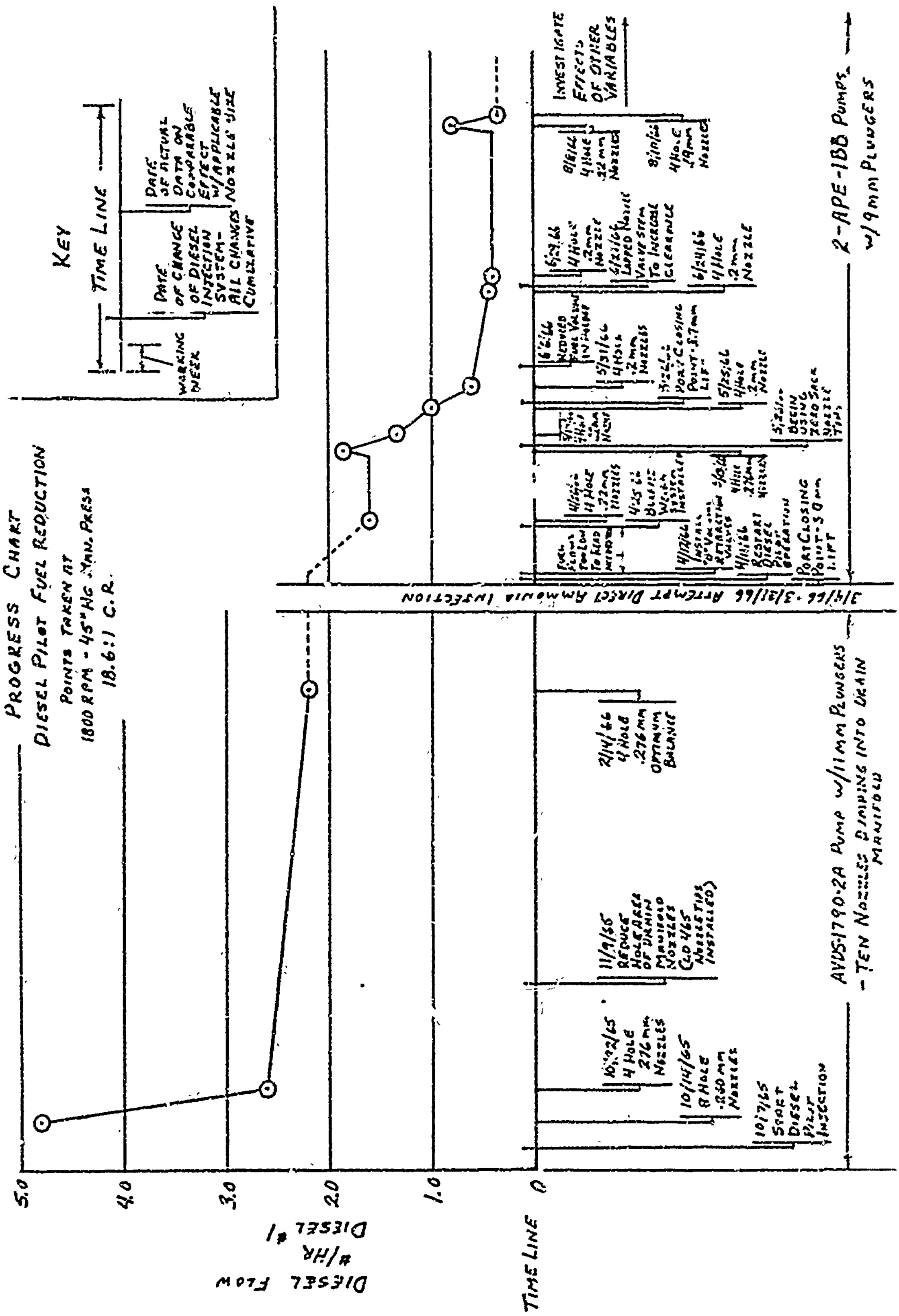

PROGRESS CHART
DIESEL PILOT FUEL REDUCTION
POINTS TAKEN AT
1800 RPM - 45" HG MAN. PRESS
18.6:1 C.R.
KEY
TIME LINE
WORKING WEEK
DATE OF CHANGE OF DIESEL INJECTION SYSTEM - ALL CHANGES CUMULATIVE
DATE OF ACTUAL DATA ON COMPARABLE EFFECT W/APPLICABLE NOZZLE SIZE
DIESEL FLOW
#/HR
DIESEL #1
5.0
4.0
3.0
2.0
1.0
0
TIME LINE
10/7/65 START DIESEL PILOT INJECTION
10/14/65 8 HOLE .250 MM NOZZLES
10/22/65 4 HOLE .276 MM NOZZLES
2/14/66 4 HOLE .276 MM OPTIMUM BALANCE
3/14/66 - 3/31/66 ATTEMPT DIRECT AMMONIA INJECTION
AVDS-1790-2A PUMP W/11 MM PLUNGERS
- TEN NOZZLES DRAINING INTO DRAIN MANIFOLD
2-APE-1BB PUMPS
W/9MM PLUNGERS
INVESTIGATE EFFECTS OF OTHER VARIABLES



Fig. 47

NH3--198

AVDS-1790 Vee-Twin

Effect of Nozzle condition on performance with Anhydrous Ammonia Vapor and DF-1 Pilot Fuel

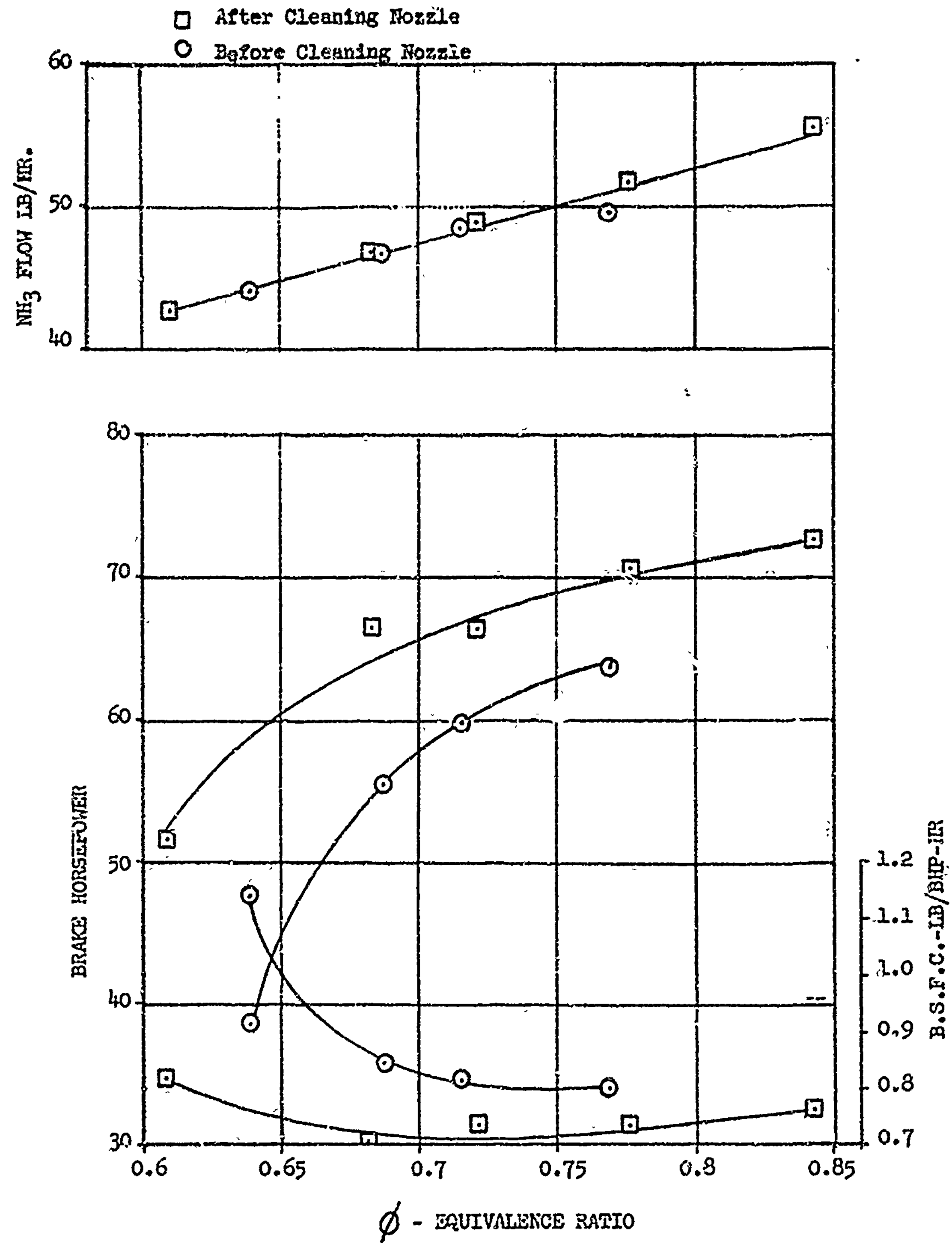





Fig. 48

NH3-190

AVDS 1790 V-Twin
Effects of Manifold Pressure on Diesel Pilot Fuel Requirements and IMEP - 1800 RPM

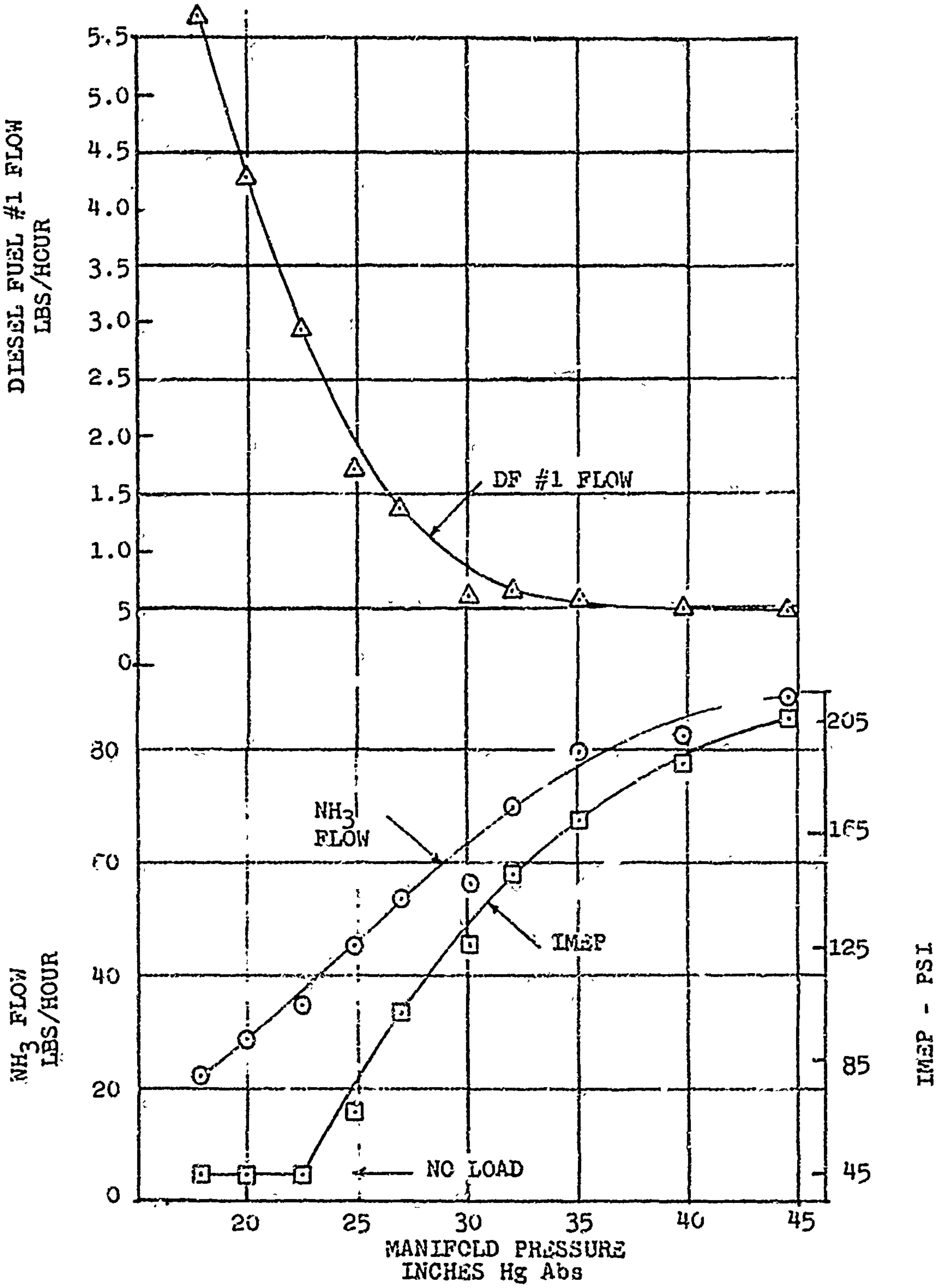





Fig. 49

INFLUENCE OF MANIFOLD PRESSURE - FUEL $NH_3$ VAPOR -
DIESEL PILOT INJECTION - V-TWIN 1790 - C.R. 18.6:1
1800 RPM

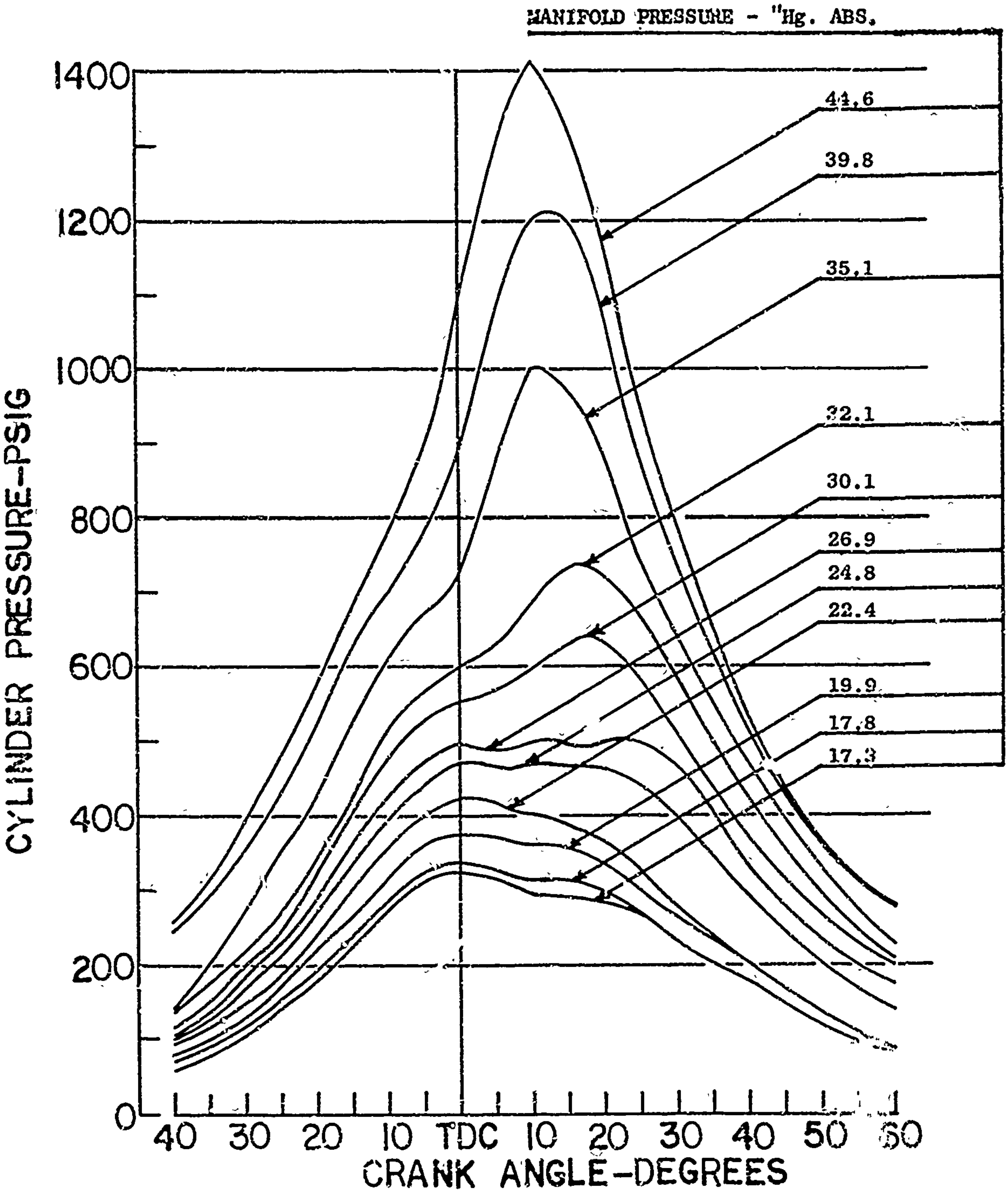



Fig. 50. Influence of Manifold Pressure on Indicator Diagrams.

## DISCUSSION

### Pilot Fuel Ignition Quality

An investigation was conducted to determine the effects of fuels having various ignition qualities. CITE fuel having a cetane number of 38, and Diesel Fuel No. 2, having a cetane number of 44, were substituted for Diesel Fuel No. 1 (cetane No. 50) as pilot fuel for igniting the ammonia-air mixture. Neither CITE fuel nor DF-2 gave near the performance obtained with DF-1. The problems were late ignition, loss of power, and inability to burn the same quantity of ammonia. With the lower ignition quality fuel, the engine demanded a much leaner fuel-air ratio

### Pilot Fuel Quantity

Experiments were conducted to determine the effects of varying the quantity of diesel pilot fuel while holding all other variables constant Figure 51 shows the results of this investigation. It is interesting to note that when operating at a desirable fuel-air ratio (0.85 equivalence ratio) the increase in output and reduction in specific fuel consumption (ammonia only) are in direct proportion to the heat content of the additional diesel pilot fuel. At a leaner fuel-air ratio (0.64 equivalence ratio), the effect of increasing the diesel pilot fuel is quite dramatic, indicating the necessity for much higher ignition energy. The increase in power output at the lower equivalence ratio is of the order of three times the heat content of the diesel fuel alone.

### Optimum Performance

The AVDS-1790 Vee-Twin was run at constant manifold pressure and constant speed, varying the fuel-air ratio to determine the optimum conditions of operation. In order to ensure good ignition and combustion and comparable performance at off-optimum points, the diesel pilot fuel was held at 4 to 5 cubic millimeters per stroke. These tests were repeated at various manifold pressures and speeds to cover the complete load and speed range of the engine. Figures 52 through 56 show the results of these tests. In general, minimum specific fuel consumption is attained at an equivalence ratio of 0.70 to 0.75; whereas maximum output is attained at an equivalence ratio of about 0.90. Increases in ammonia flow beyond this point, actually result in a loss in power.

Figure 57 shows the envelopes of indicated specific fuel consumption versus indicated horsepower for each of the various speeds. As would be

AVDS-1790

EFFECT OF VARIATION OF PILOT FUEL QUANTITY AT 1500 RPM AND TWO EQUIVALENCE RATIOS

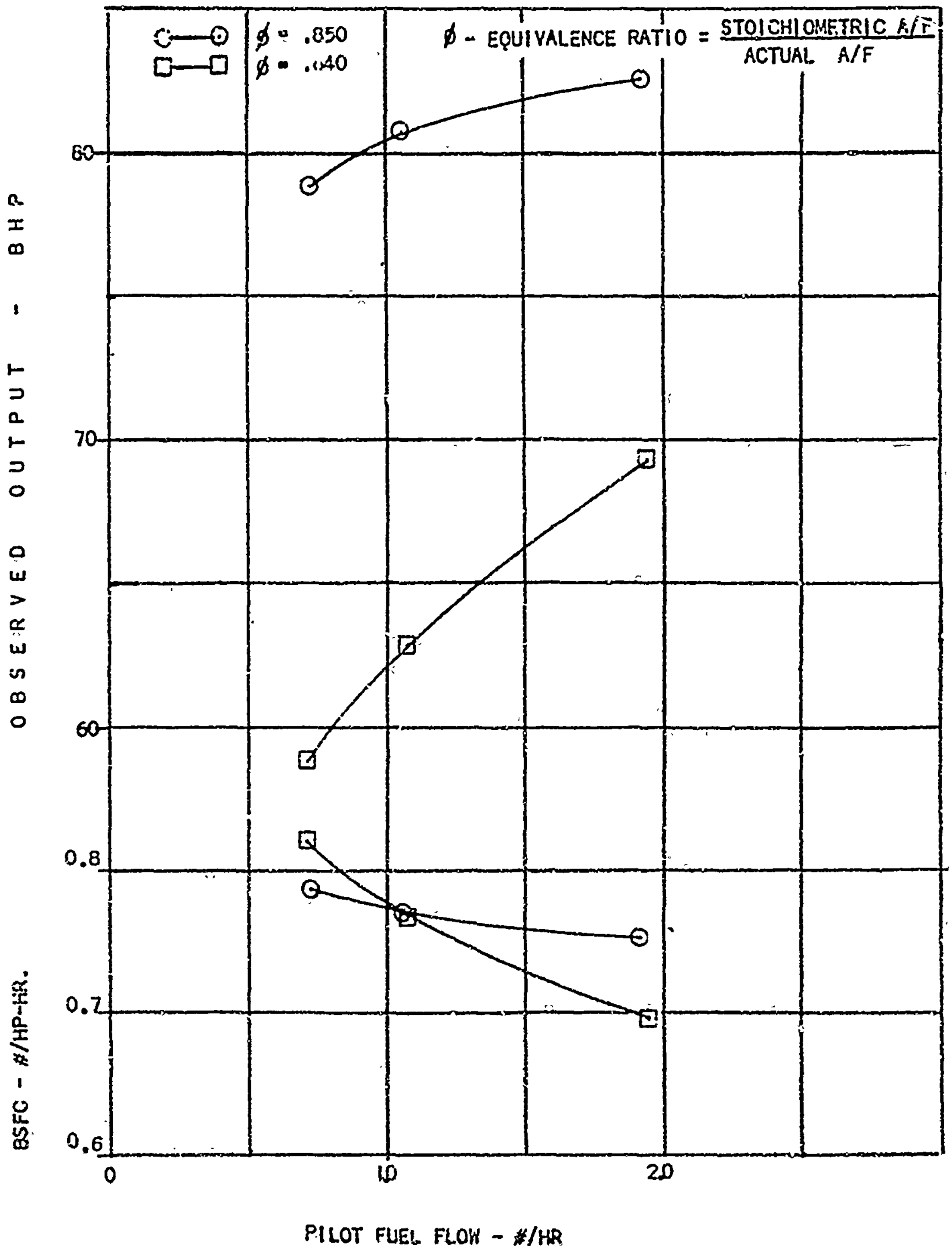





Fig. 51

NH3 - 203

AVDS-1790 V-Twin #3
ISFC and Equivalence Ratio Vs. Indicated Horsepower
$NH_3$ Vapor and Diesel Pilot Fuel @ 1200 RPM

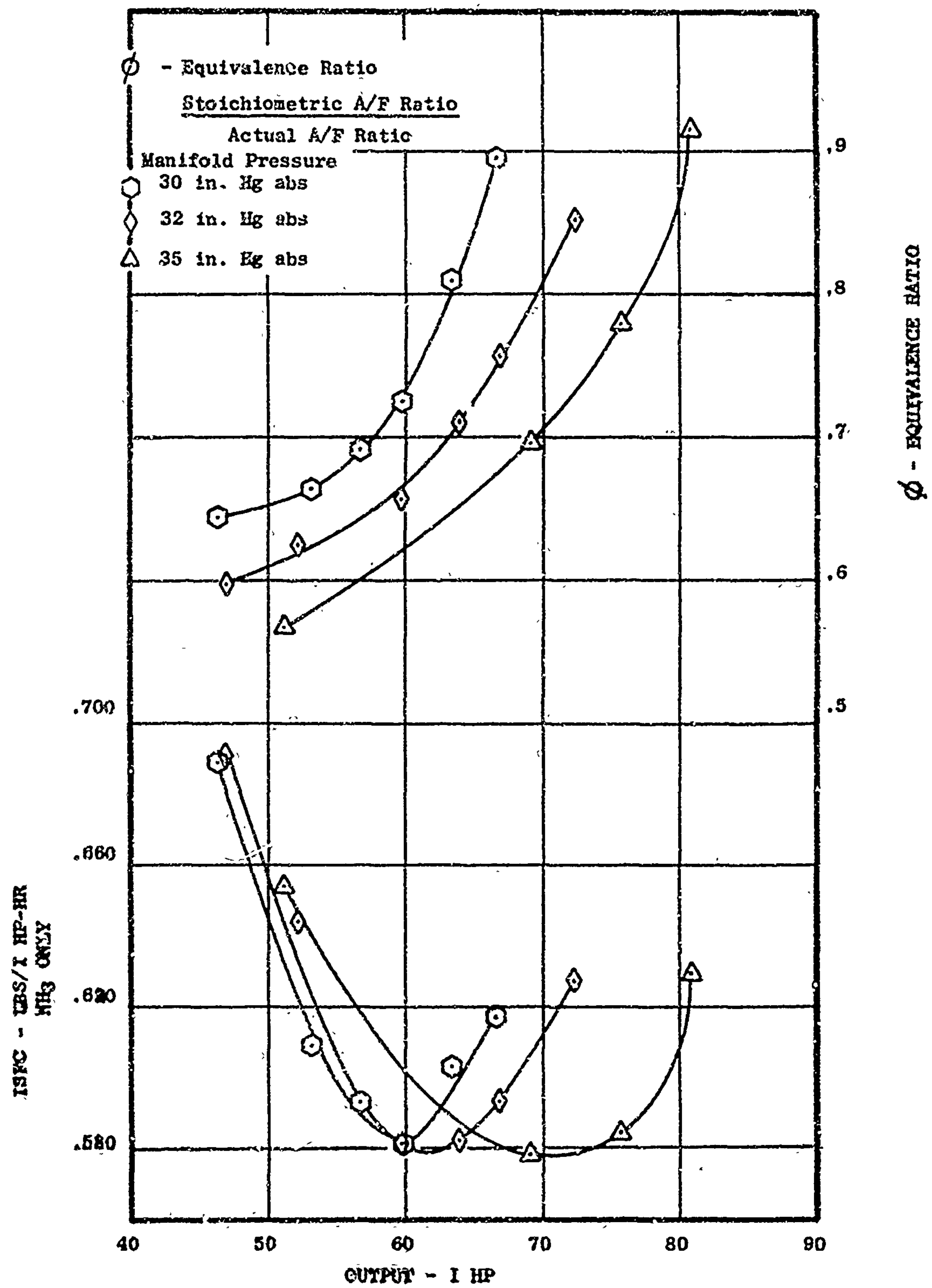





Fig. 52

AVDS-1790 V-Twin #3
ISFC and Equivalence Ratio Vs. Indicated Horsepower
$NH_3$ Vapor and Diesel Pilot Fuel @ 1500 RPM

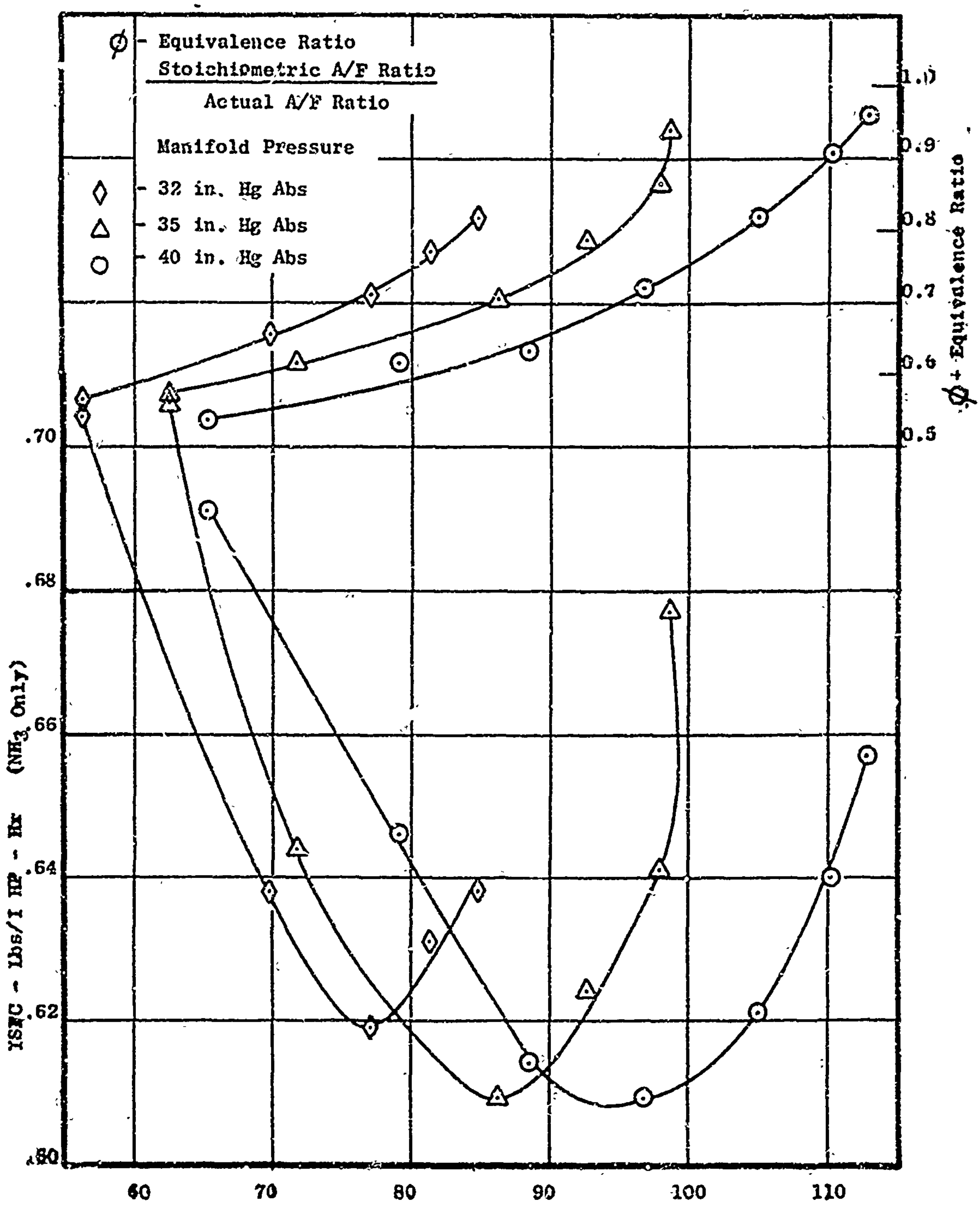





Fig. 53

NH3- 204

AVDS-1790 V-Twin #3

ISFC and Equivalence Ratio Vs. Indicated Horsepower
NH3 Vapor and Diesel Fuel @ 1800 RPM

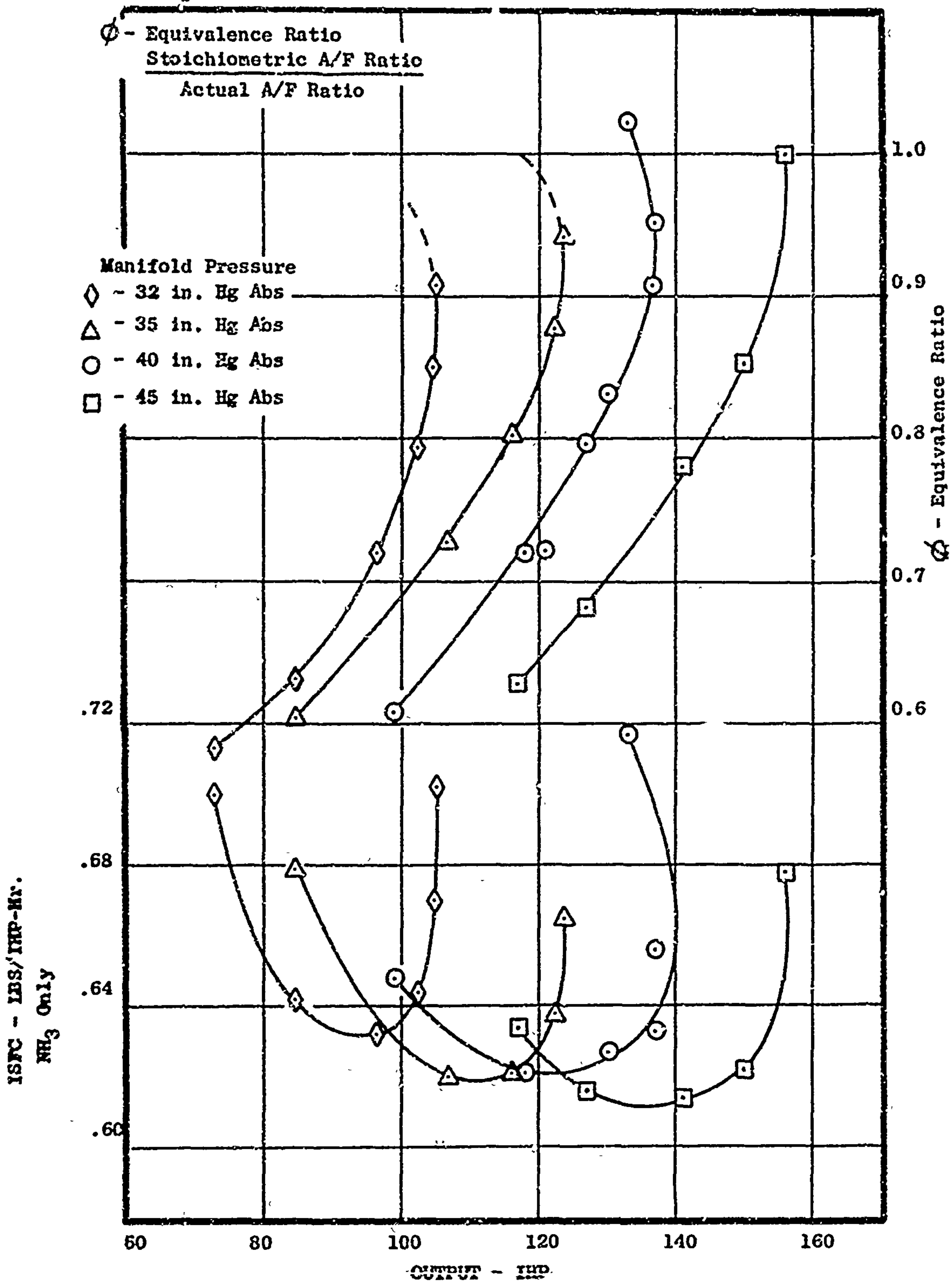
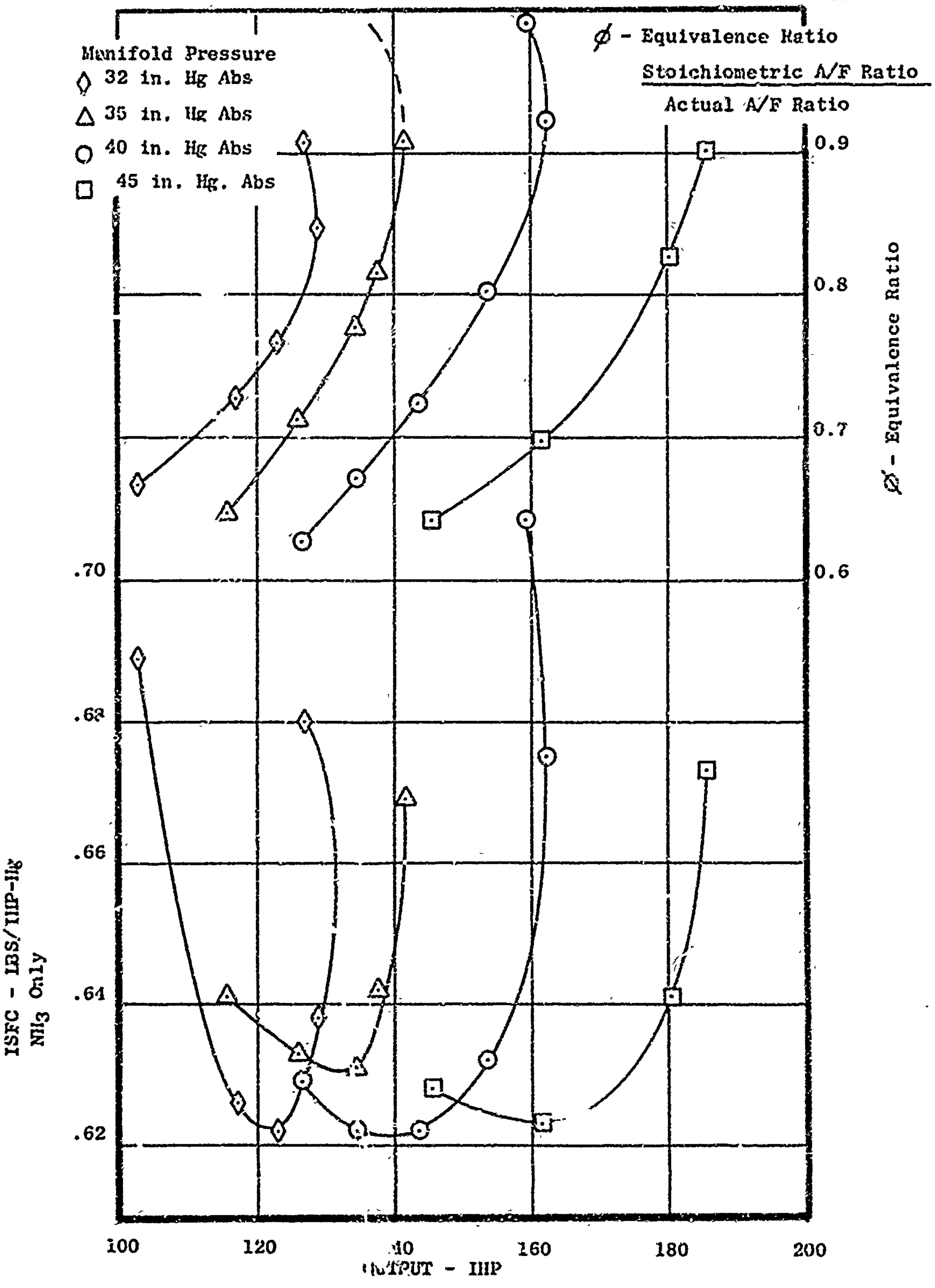



Fig. 54

AVDS-1790 V-Twin #3

ISFC and Equivalence Ratio Vs. Indicated Horsepower
$NH_3$ Vapor and Diesel Pilot Fuel @ 2100 RPM




764-784, 834-876



Fig. 55

$NH_3$ - 200

AVDS 1790 V-Twin #3

Indicated Horsepower Vs. ISFC and Equivalence Ratio Operated on $NH_3$ Vapor and Diesel Pilot Injection @ 2400 RPM

H, VanSweden 12-23-66 Nov. 66

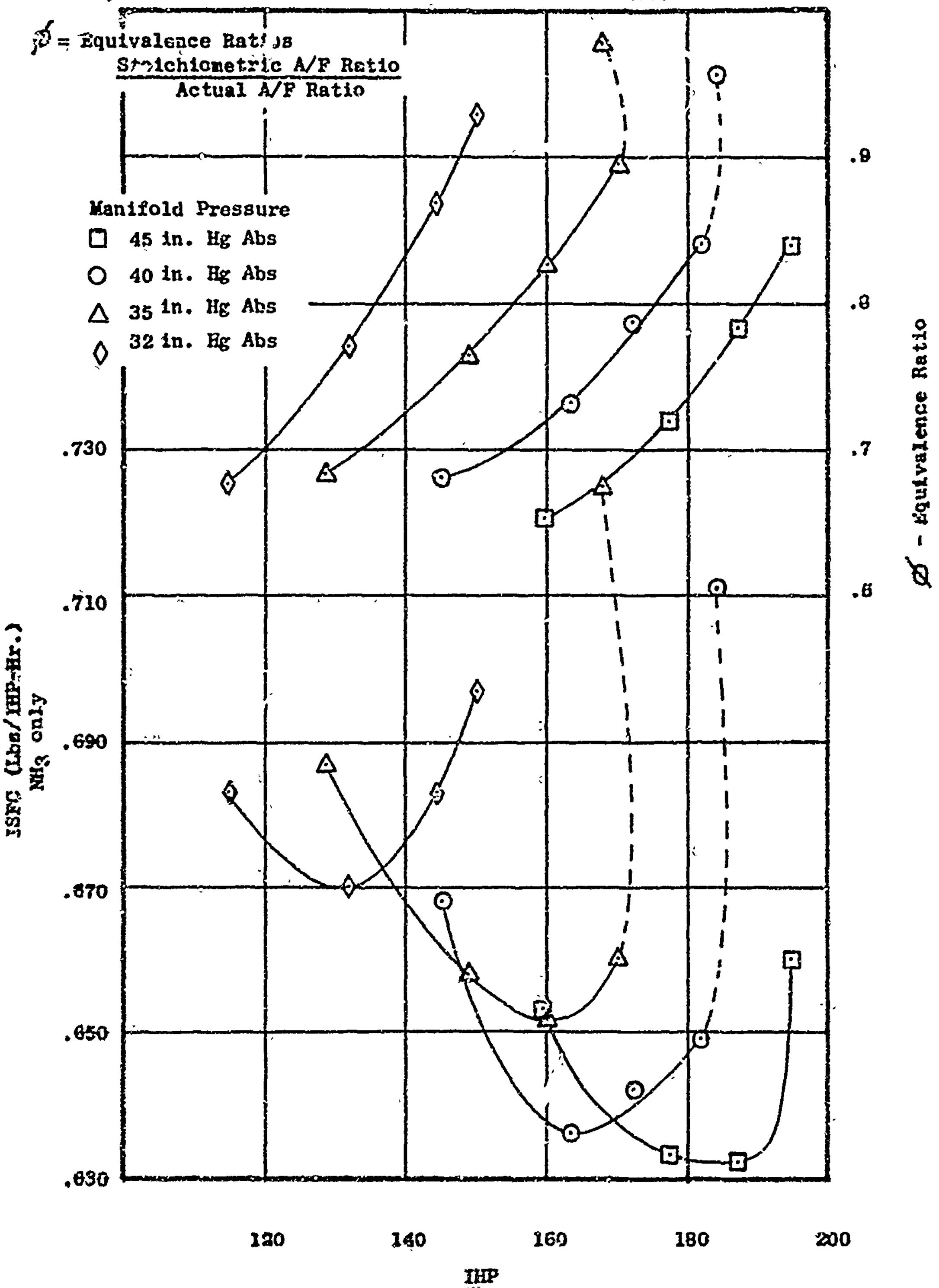





Fig. 56

$NH_3$ - 240

#703 1,90 V-Twin #3

Part Load Performance ISFC vs IHP on $NH_3$ Vapor with Diesel Pilot

Injection Quantities Approximate 4-5 MM 3/Stroke 18.6:1 CR

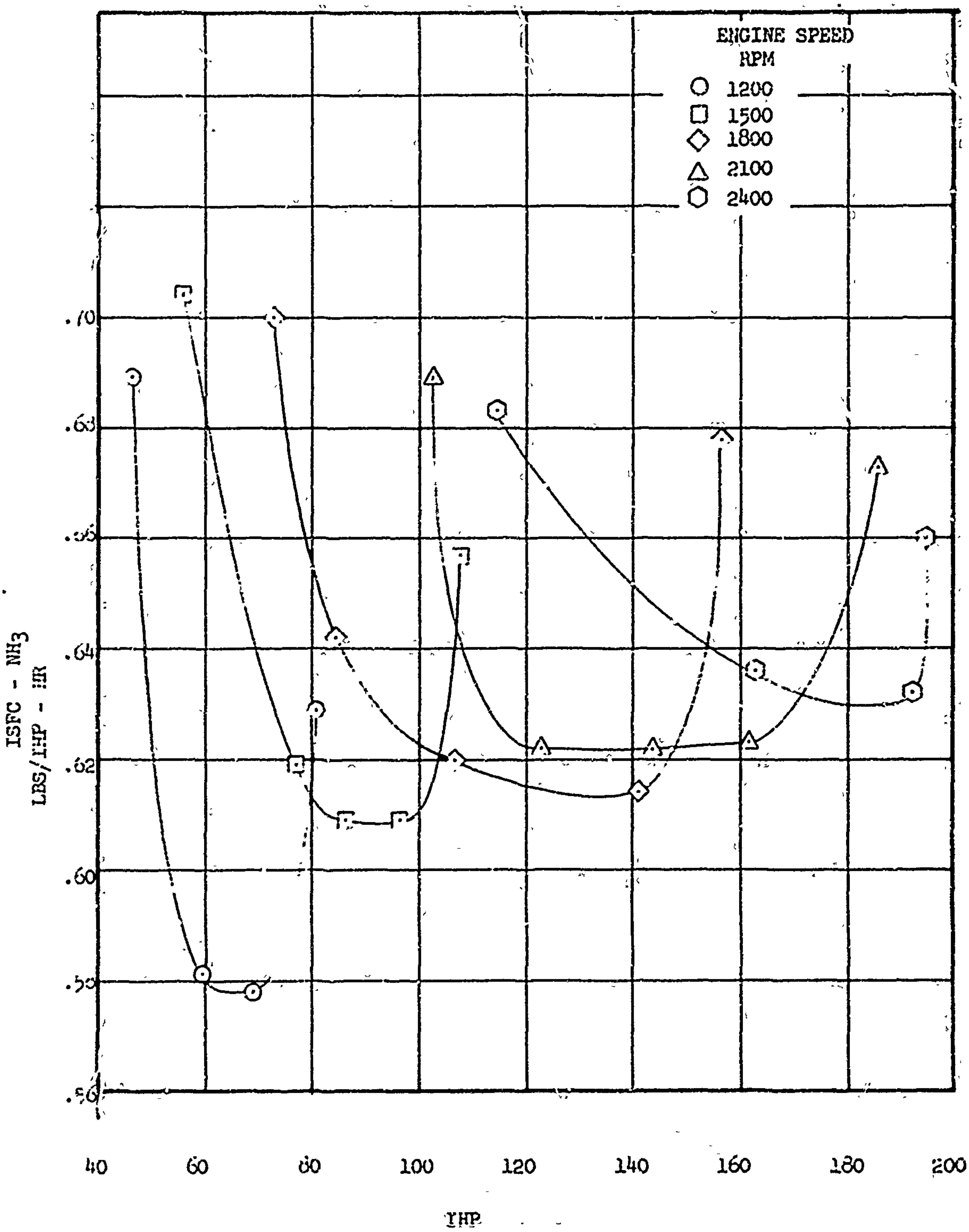



Fig. 57

DISCUSSION

expected, the best fuel consumption occurs at 1200 rpm and the worst at 2400 rpm. This is considered to be a reflection of the effect of the low flame propagation rate of ammonia-air mixtures.

Spark-Ignition

The AVDS-1790 Vee-Twin was converted to a spark-ignition engine by doing the following:

1. Removing the fuel injection pumps and installing an eight-cylinder Mallory "Super-Mag" Magneto. An eight-cylinder magneto was chosen in order to be able to obtain the 270° - 450° firing interval. The excess six leads in the magneto were grounded out.

2. The cylinders were removed and the holes for the fuel injection nozzles were reworked to accept spark plugs. 12mm spark plugs were used as a matter of convenience and to minimize machining costs.

Three compression ratios (18.6:1, 16:1 and 12:1) were investigated in the spark-ignition studies. Figures 58 and 59 show the combustion chamber shapes and the relative location of the fuel injection nozzles and the spark plugs. It should be noted that the longest reach spark plugs, consistent with piston clearance, were installed. No additional development was undertaken in this phase of the work; the best approaches developed under the L-141 program were applied directly to this engine.

Spark Plug Gap

At each of the three compression ratios investigated, the spark plug gaps were varied to determine the optimum setting. Figure 60 shows the effect of spark plug gap on brake specific fuel consumption. As is to be expected, the optimum gap varies inversely with compression ratio. The optimum gap at 12:1 compression ratio, 0.070-inch, is considerably less than the 0.100-inch at 12.6:1 compresstion ratio required by the L-141 engine. There is no apparent explanation for this difference.

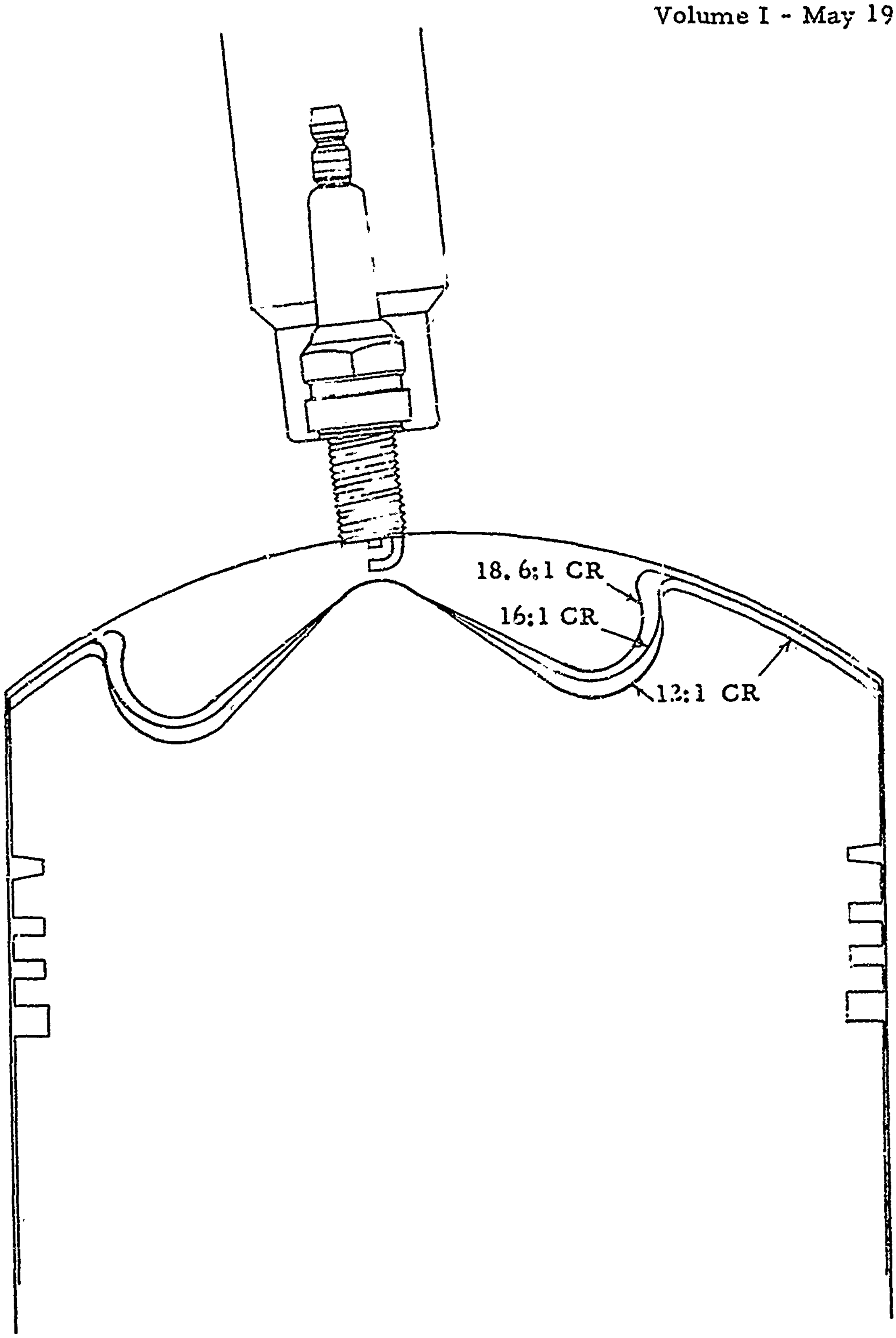



Fig. 58. Sketch of 18.6:1 Compression Ratio Combustion Chamber Showing Location of Fuel Injection Nozzle.

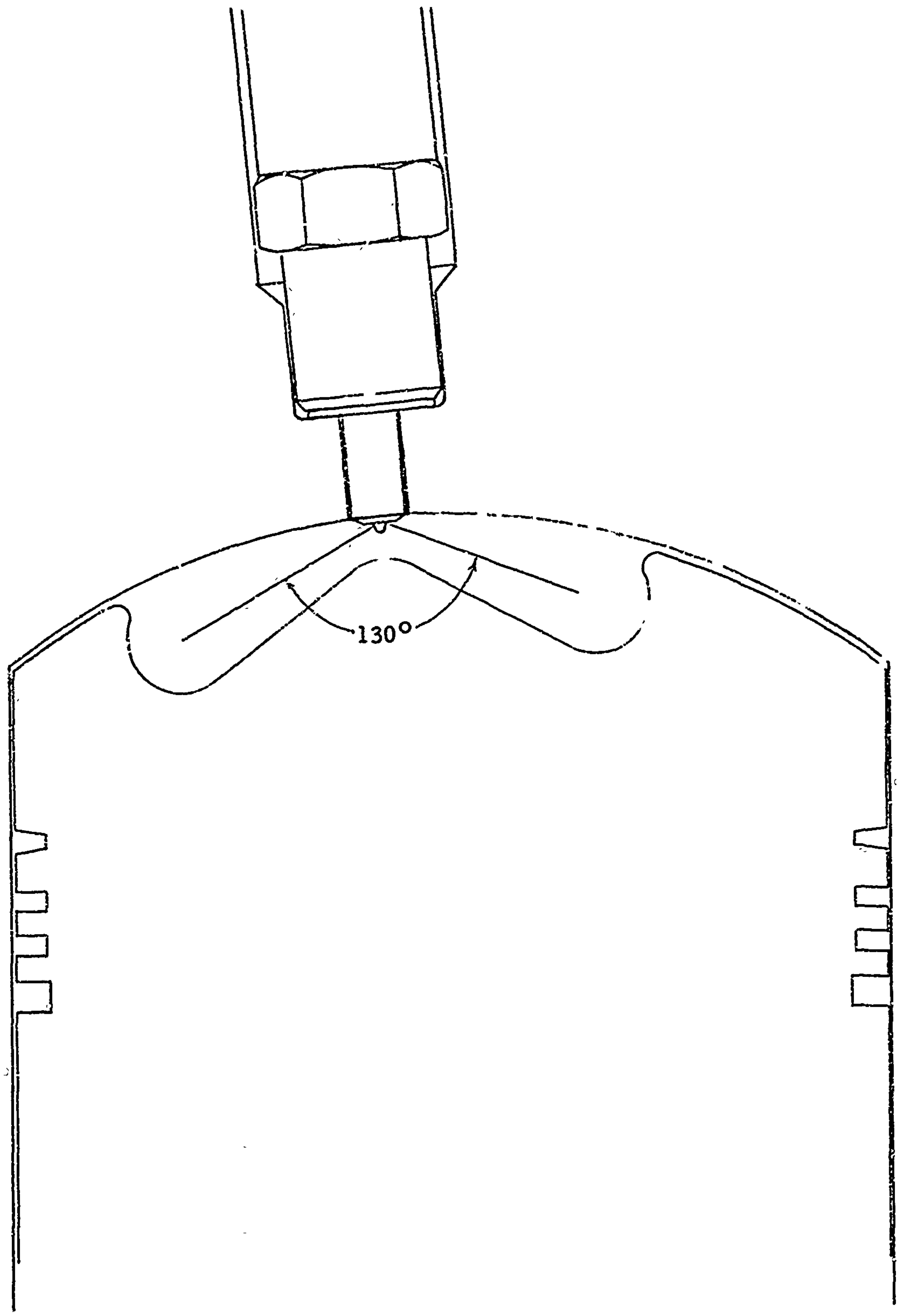



Fig. 59. Sketch of Spark-Ignition Combustion Chamber Showing Changes With Various Compression Ratios and Location of Spark Plug.

AVDS-1790 V-Twin #3
Effect of Spark Plug Gap on BSFC at Three Compression Ratios.
Mallory Magneto, Champion R-Series Plugs, 12 MM 3/4" Reach

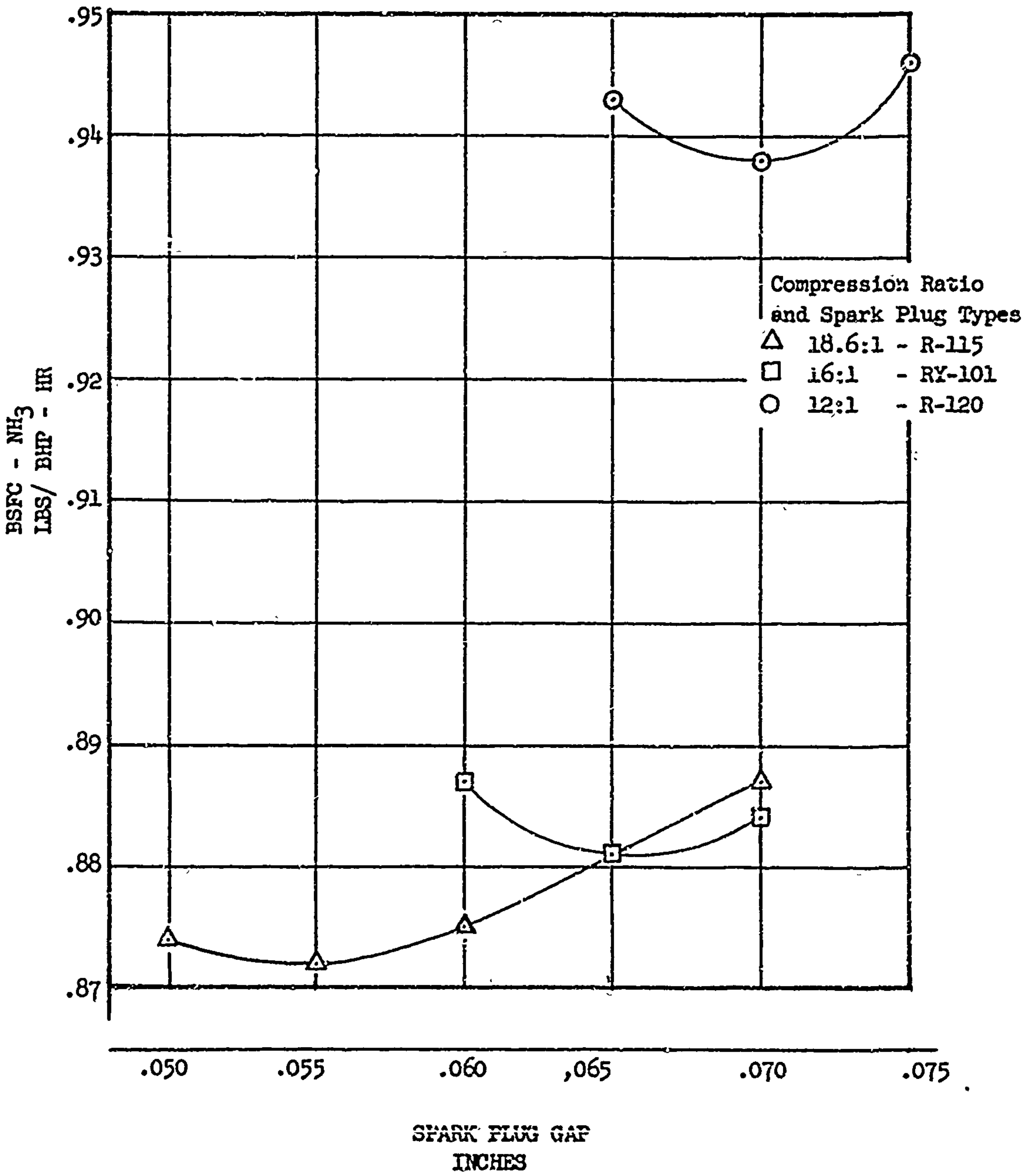



Fig. 60

# DISCUSSION

## Spark Plug Durability

The spark plugs used in this investigation were designed for use in conventional Otto cycle engines and were not intended for operation with the high firing pressures encountered in this diesel engine conversion. When running at 18.6:1 compression ratio and 32 inches of mercury manifold pressure, peak pressures of 1450 psi were experienced. As a result of these pressures there were some failures due to internal gas leakage and to acutal physical movement of the center electrode and it is porcelain insulation. Figure 61 shows two plugs that failed in this manner compared with an undamaged plug. For this reason further testing was limited to naturally aspirated operation and no more failures were encountered.

## Compression Ratio

Figures 62 through 74 show specific fuel consumption and equivalence ratio at 18.6:1, 16:1 and 12:1 compression ratio for speeds of 1200 to 2400 rpm. Reducing the compression ratio from 18.6:1 to 16:1 had slight effect on specific fuel consumption; it did, however, reduce peak firing pressures by approximately 100 psi. Dropping the compression ratio still further to 12:1 did effect fuel consumption materially, particularly in the high speed range, where an increase of six percent was observed. Figure 75 shows the variation in firing pressures and specific fuel consumption with compression ratio at 1800 rpm.

## Performance Comparisons

Direct comparison of performance between operation using pilot fuel for ignition and operation using spark-ignition is difficult because of the different mode of testing. The compression-ignition engine using pilot fuel could not be run with throttling of the induction air because of high ignition lag. The spark-ignition engine, on the other hand, could not be operated supercharged because of the limitations on available spark plugs. Close examination of the data, however, does reveal the following trends:

1. With spark-ignition, actual ignition of the fuel begins about 10 degrees earlier than with pilot ignition.

2. With spark-ignition, peak cylinder pressure is about 500 psi higher and the peak pressure occurs about seven degrees earlier.

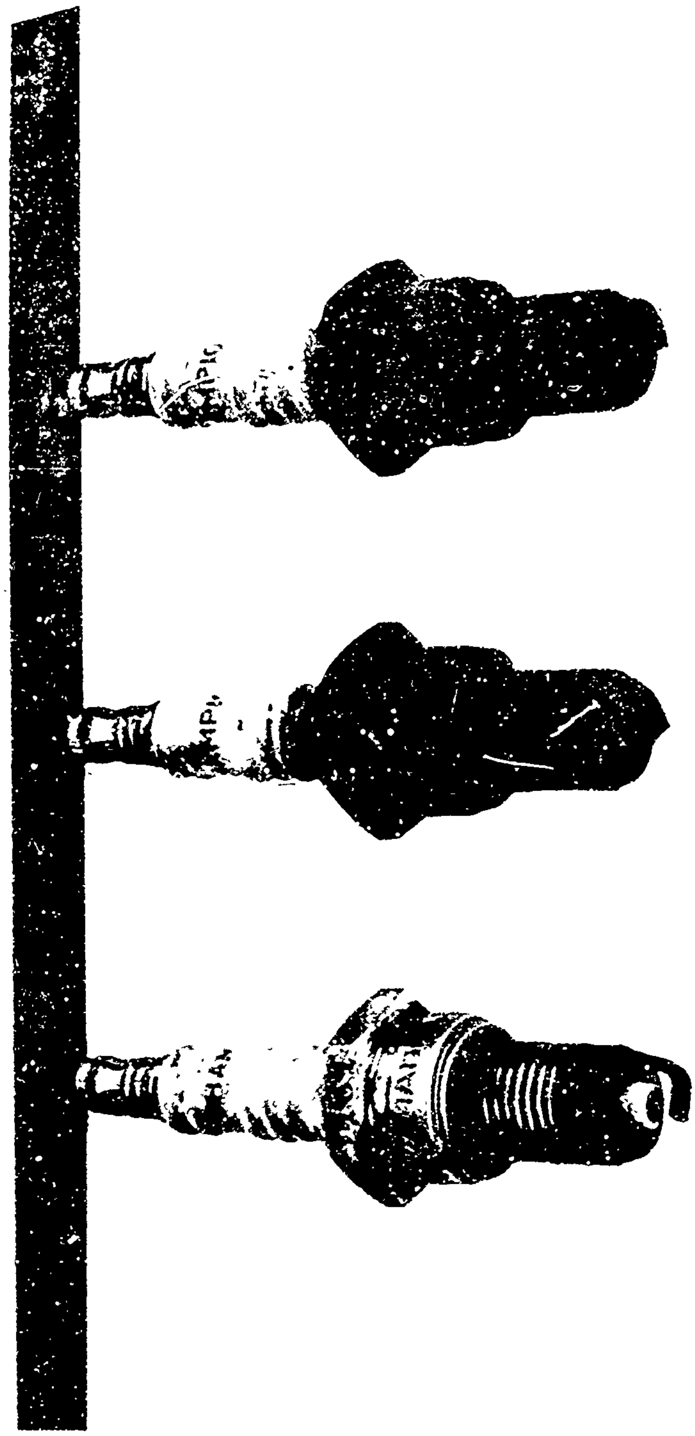


Fig. 61. Two Used Spark Plugs Versus New Spark Plug Showing Movement of Center Electrode and Porecelain Insulation. (D-35548)

AVDS 1790 V-Twin #3
ISFC and Equivalent Ratio Vs Indicated Horsepower on $NH_3$ Vapor with Mallory Mag., R-115 Plugs, Gap .055", 18.6:1 CR @ 1200 RPM

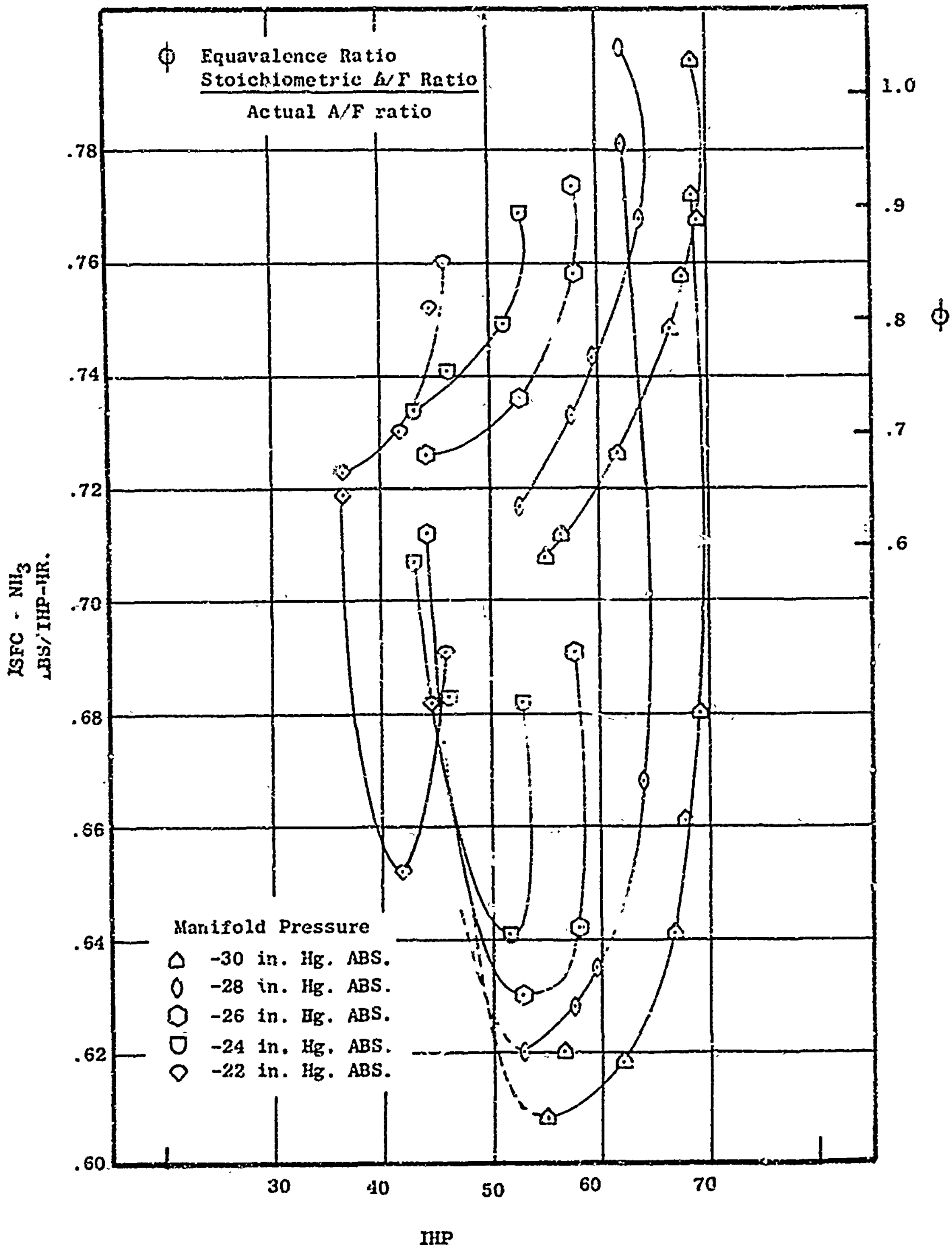



Fig. 62

AVDS 1790 V-Twin #3

ISFC and Equivalence Ratio Vs. Indicated Horsepower on $NH_3$ Vapor with Mallory Mag., R115 Plugs, Gap.055" 18.6:1 CR @ 1500 RPM

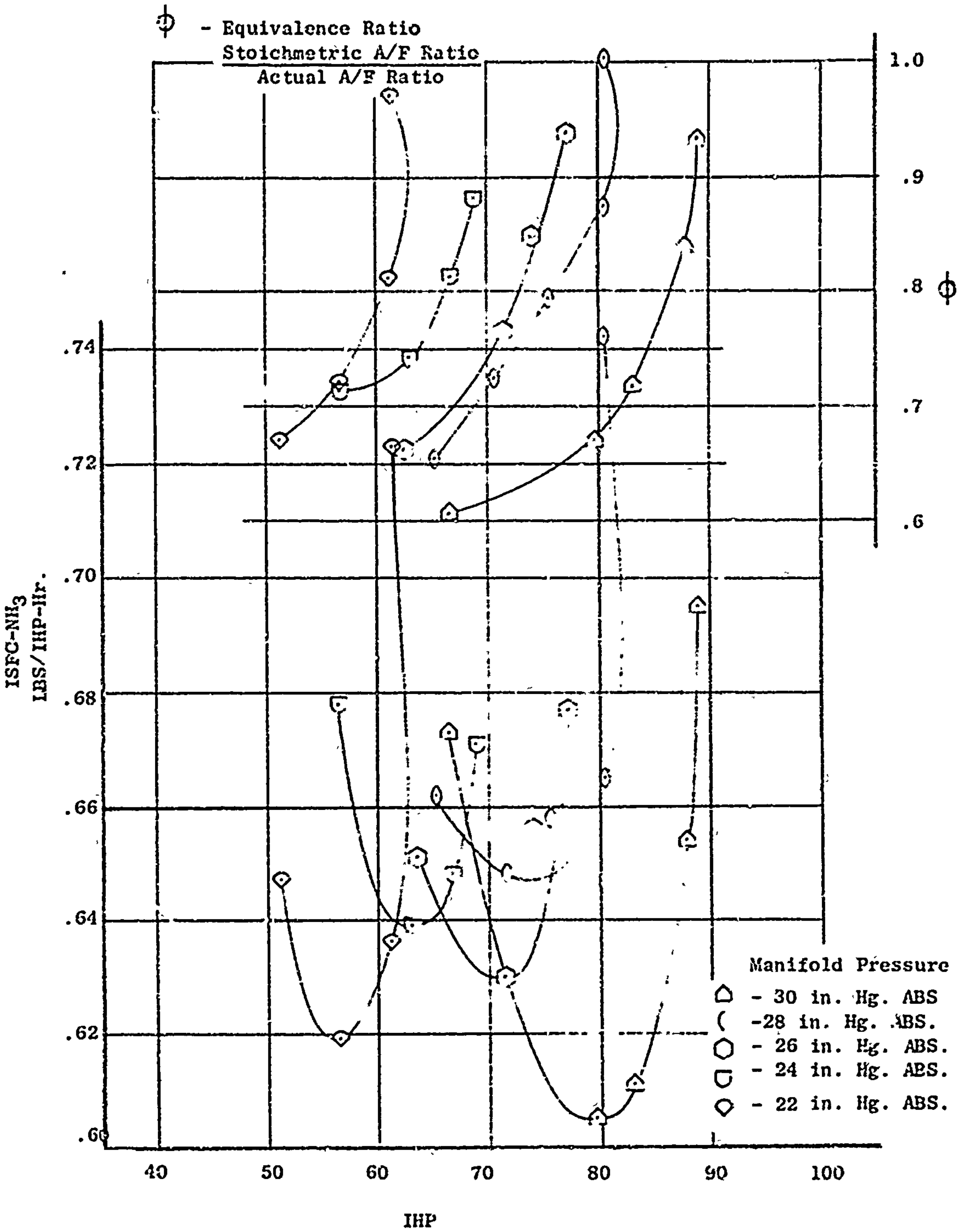





Fig. 63

AVDS 1790 V-Twin #3
ISFC and Equivalence Ratio Vs. Indicated Horsepower on NH3 Vapor with Mallory Mag., R-115 Plugs, Gap .055" 18:6:1 CR @ 1800 RPM

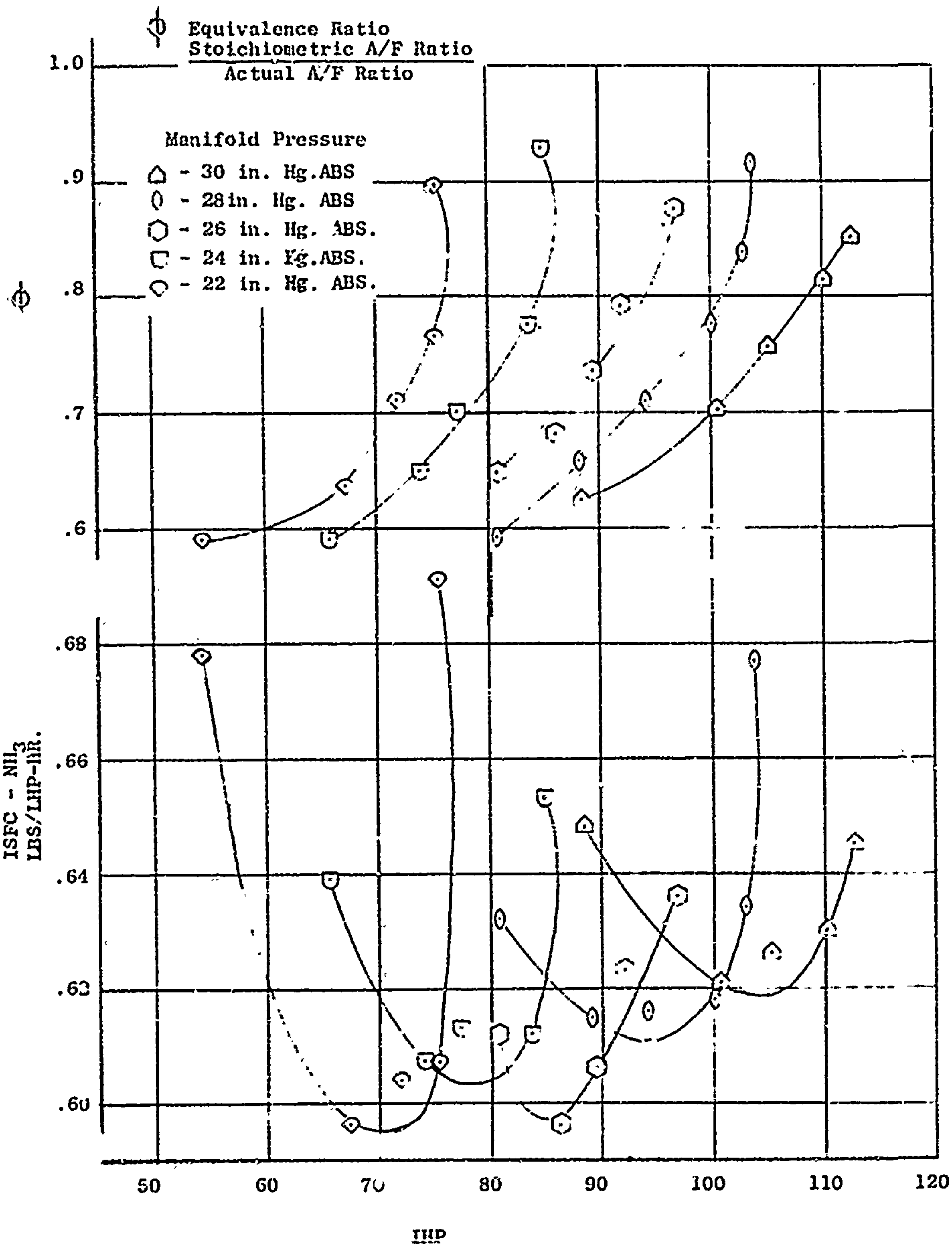





Fig. 64

$NH_3$ - 225

AVDS 1790 V-Twin #3

ISFC and Equivalence Ratio Vs Indicated Horsepower on $NH_3$ Vapor with Mallory Mag. R-116 Plugs, Gap .055" 78.6:1 CR @ 2100 RPM

H. VanSweden

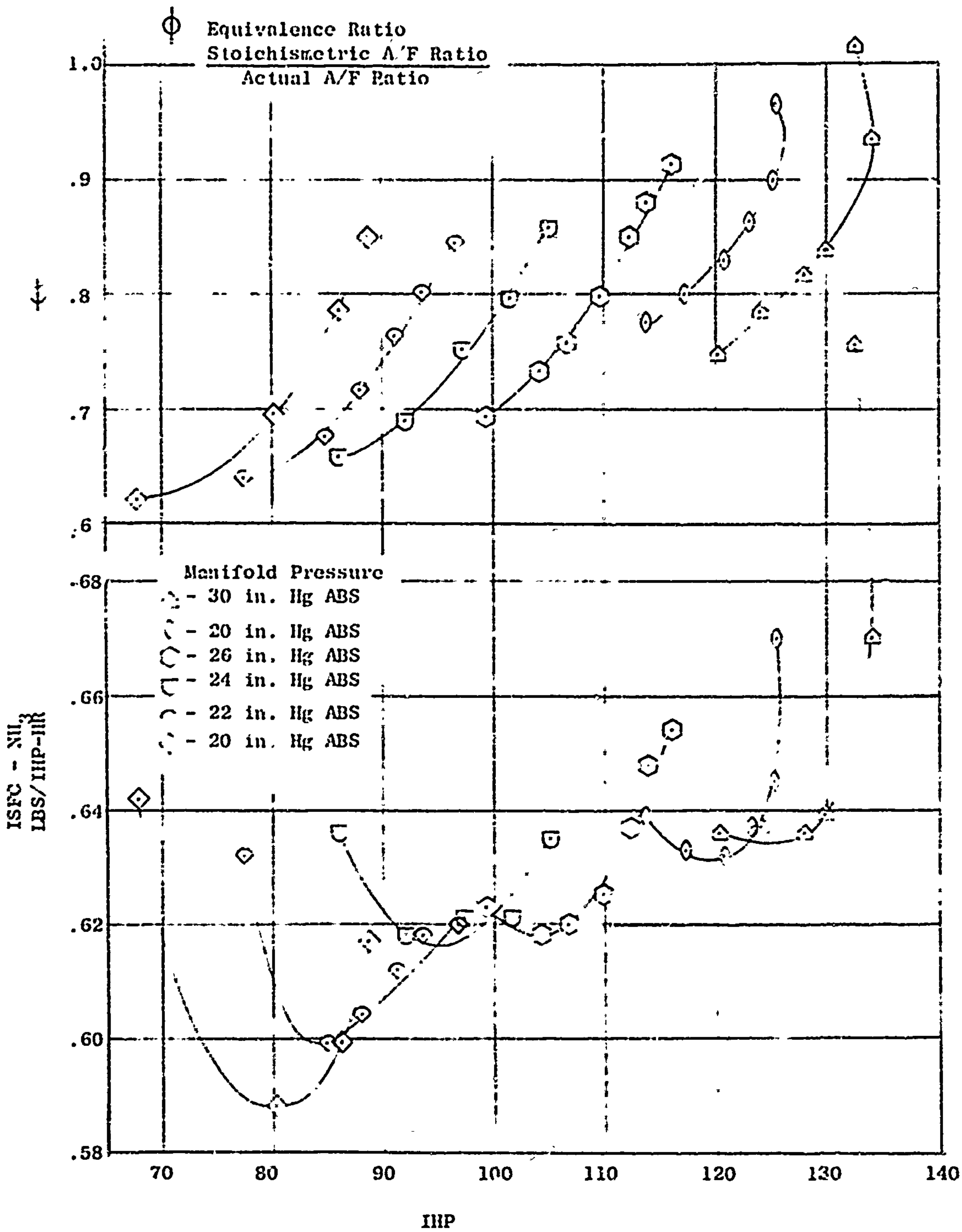



Fig. 65

AVDS 1790 V-Twin #3
ISFC and Equivalence Ratio Vs. Indicated Horsepower on $NH_3$ Vapor with Mallory Mag. R-116 Plugs, Gap .055" 18.6:1 CR @ 2400 RPM

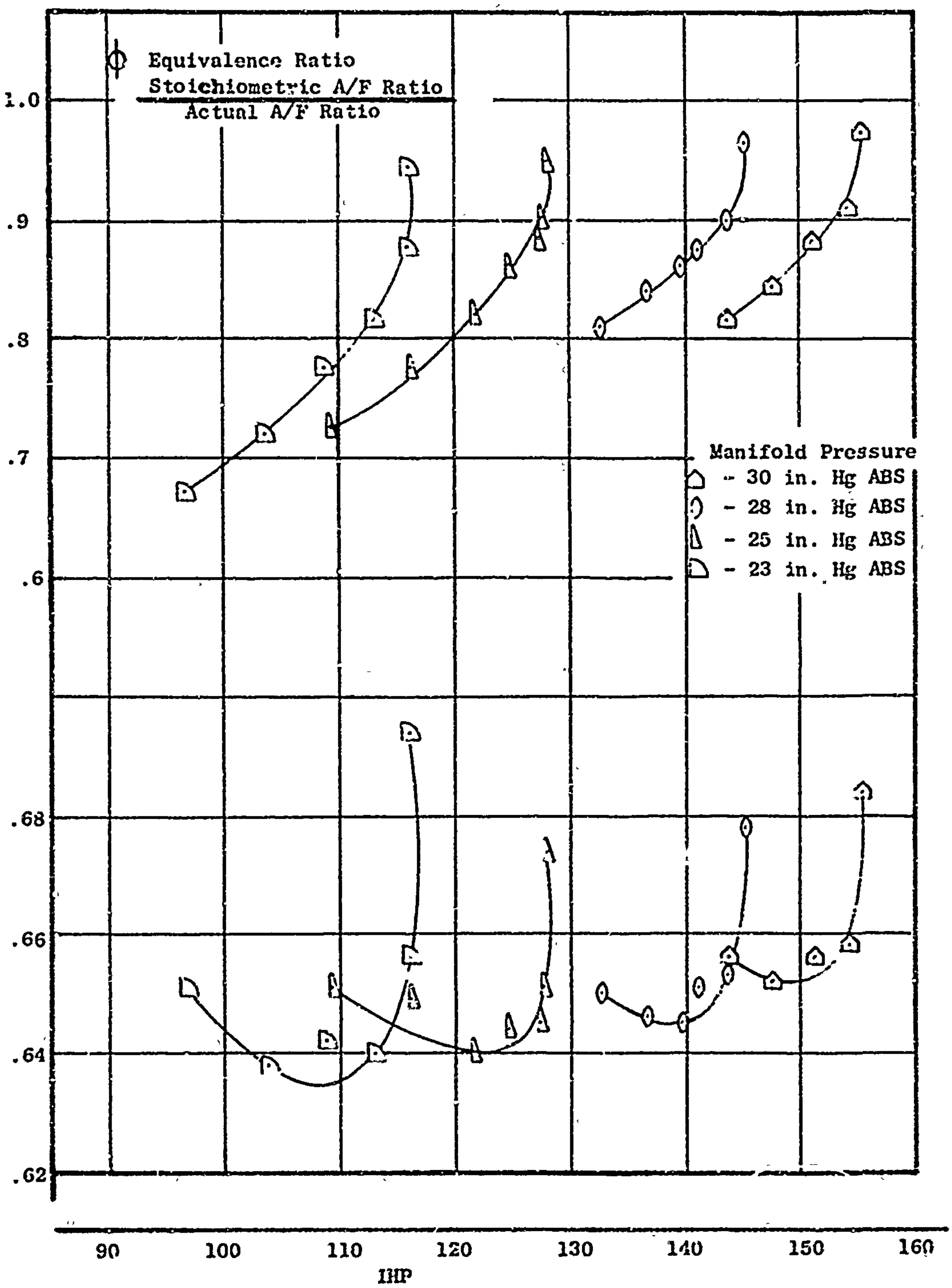



Fig. 66

AVDS-1790 V-Twin

ISFC and Equivalence Ratio vs. Indicated Horsepower on $NH_3$ Vapor with Mallory Mag., R-120 Plugs, Gap .065", 16:1 CR @ 1200 RPM

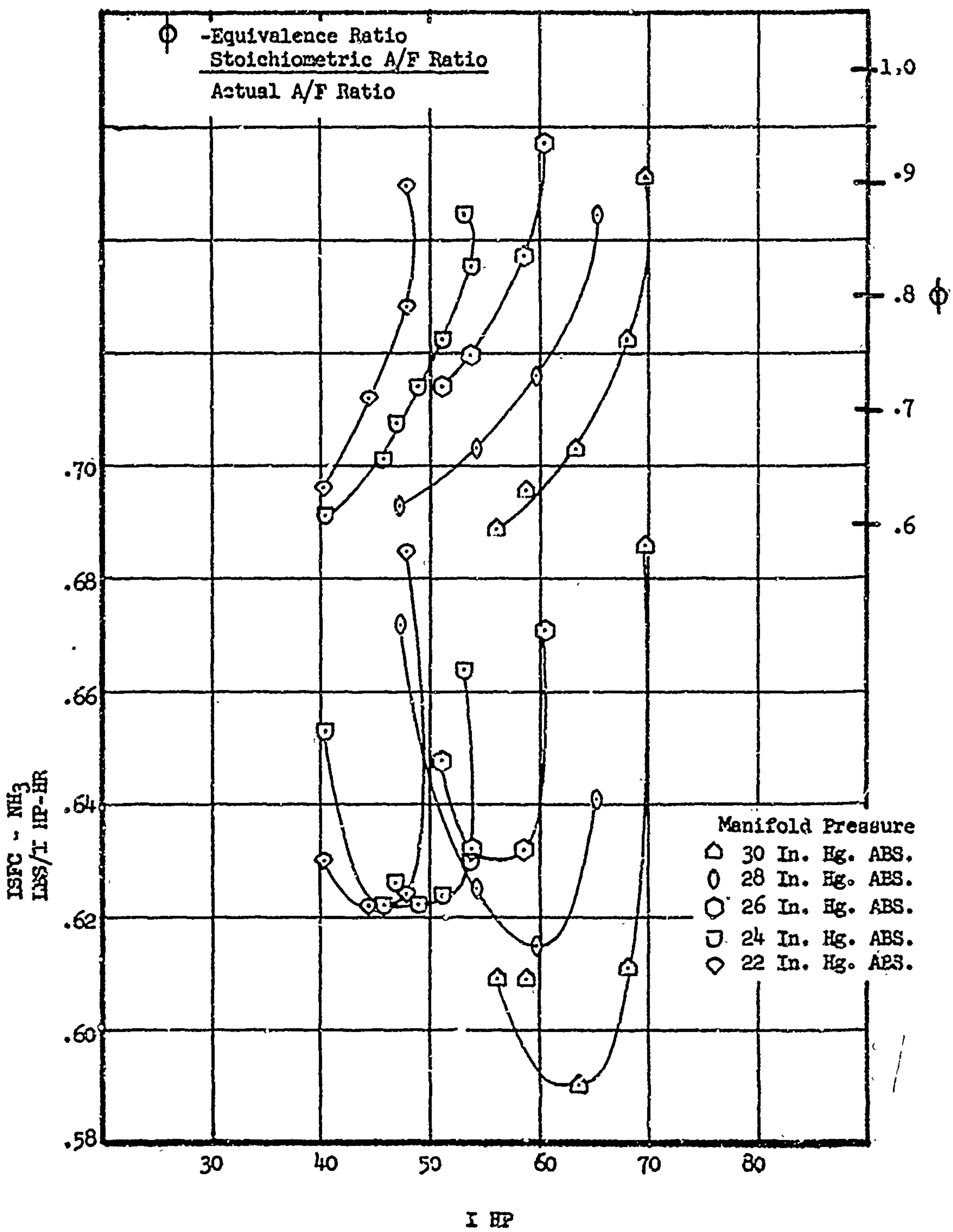





Fig. 67

NH3 - 228

AVDS-1790 V-Twin #3

ISFC and Equivalence Ratio vs. Indicated Horsepower on $NH_3$ Vapor with Mallory Mag., R-120 Plugs, Gap .065", 16:1 CR @ 1800 RPM

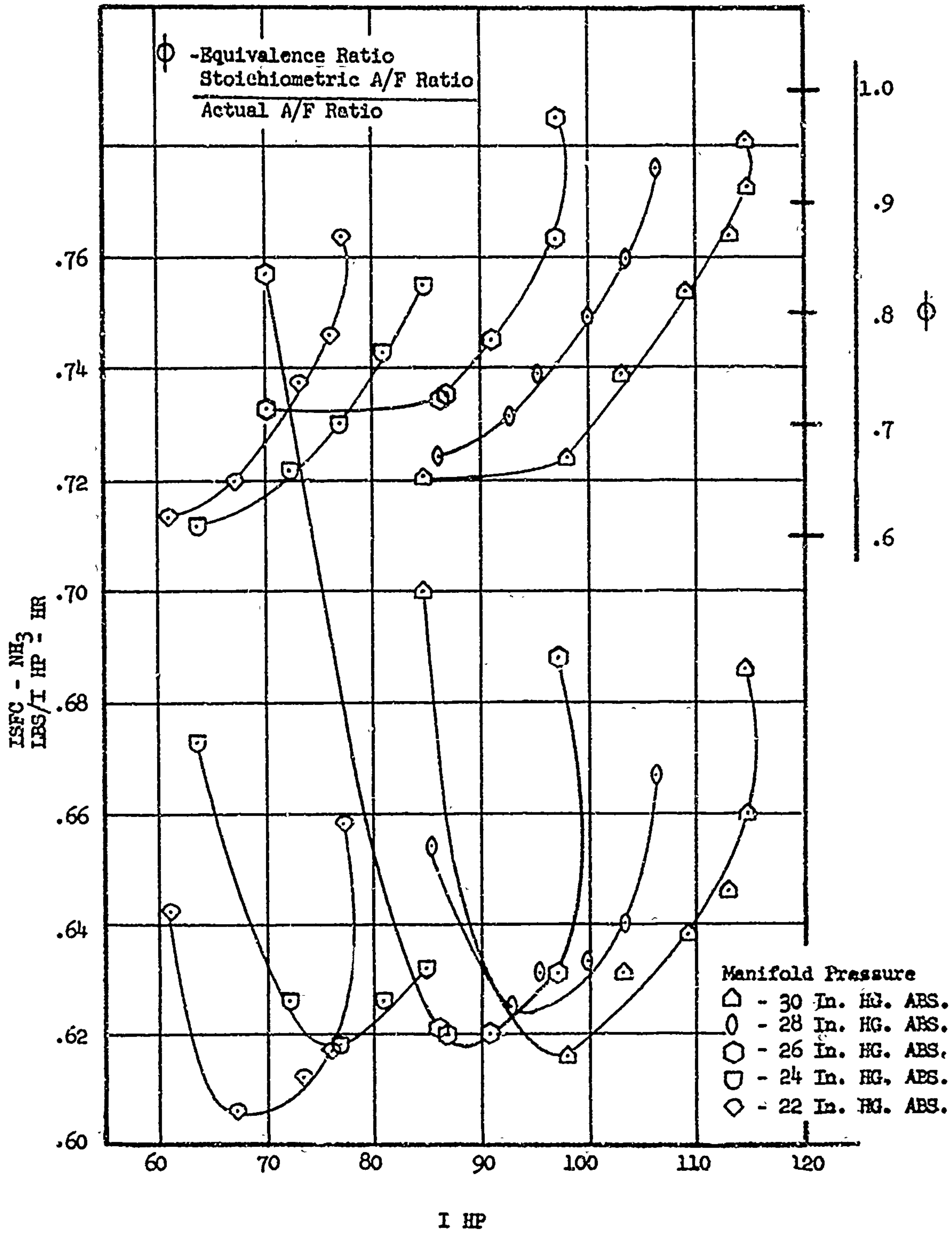





Fig. 68

AVDS-1790 V-Twin #3

ISFC and Equivalence Ratio vs. Indicated Horsepower on $NH_3$ Vapor with Mallory Mag., R-120 Plugs, Gap .065", 16:1 CR @ 2400 RPM

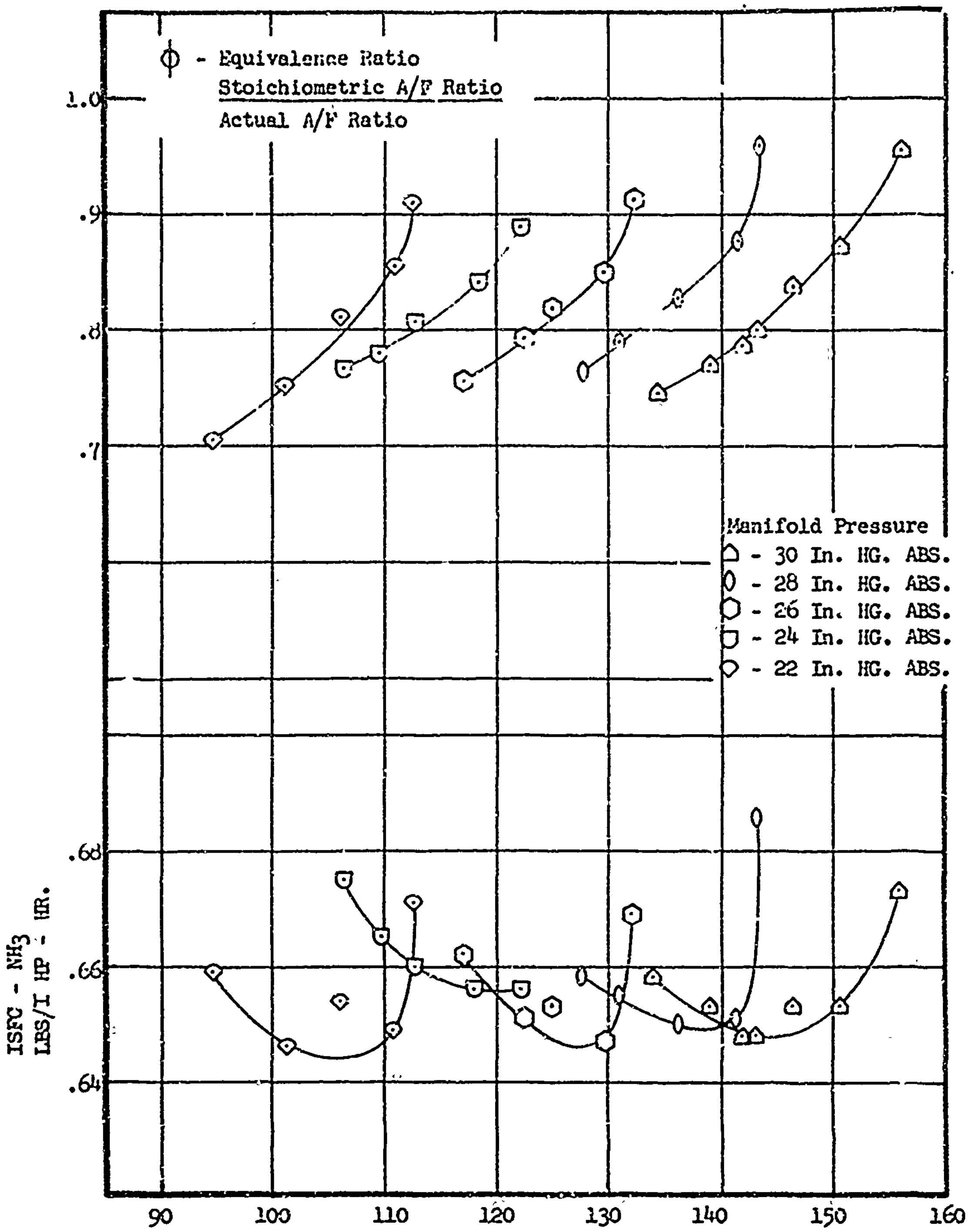





Fig. 69

NH3 - 230

AVDS-1790 V-Twin #3

ISFC and Equivalence Ratio vs. Indicated Horsepower on $NH_3$ Vapor with Mallory Mag., R-120 Plugs, Gap .070", 12:1 CR @ 1200 RPM

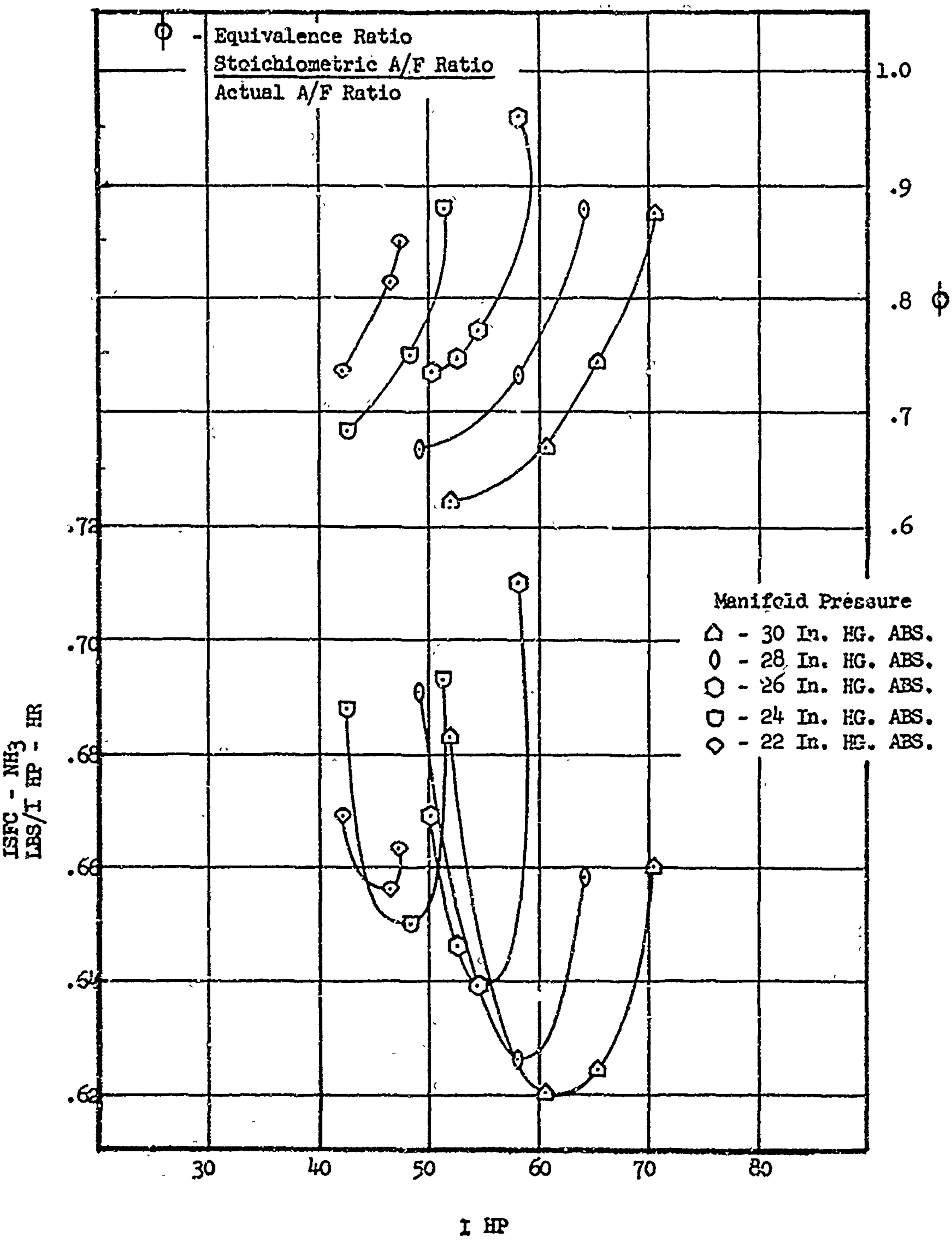





Fig. 70

AVDS-1790 V-Twin #3

ISFC and Equivalence Ratio vs. Indicated Horsepower on $NH_3$ Vapor with Mallory Mag., R-120 Plugs, Gap .070", 12:1 CR @ 1500 RPM

H. Van Sweden

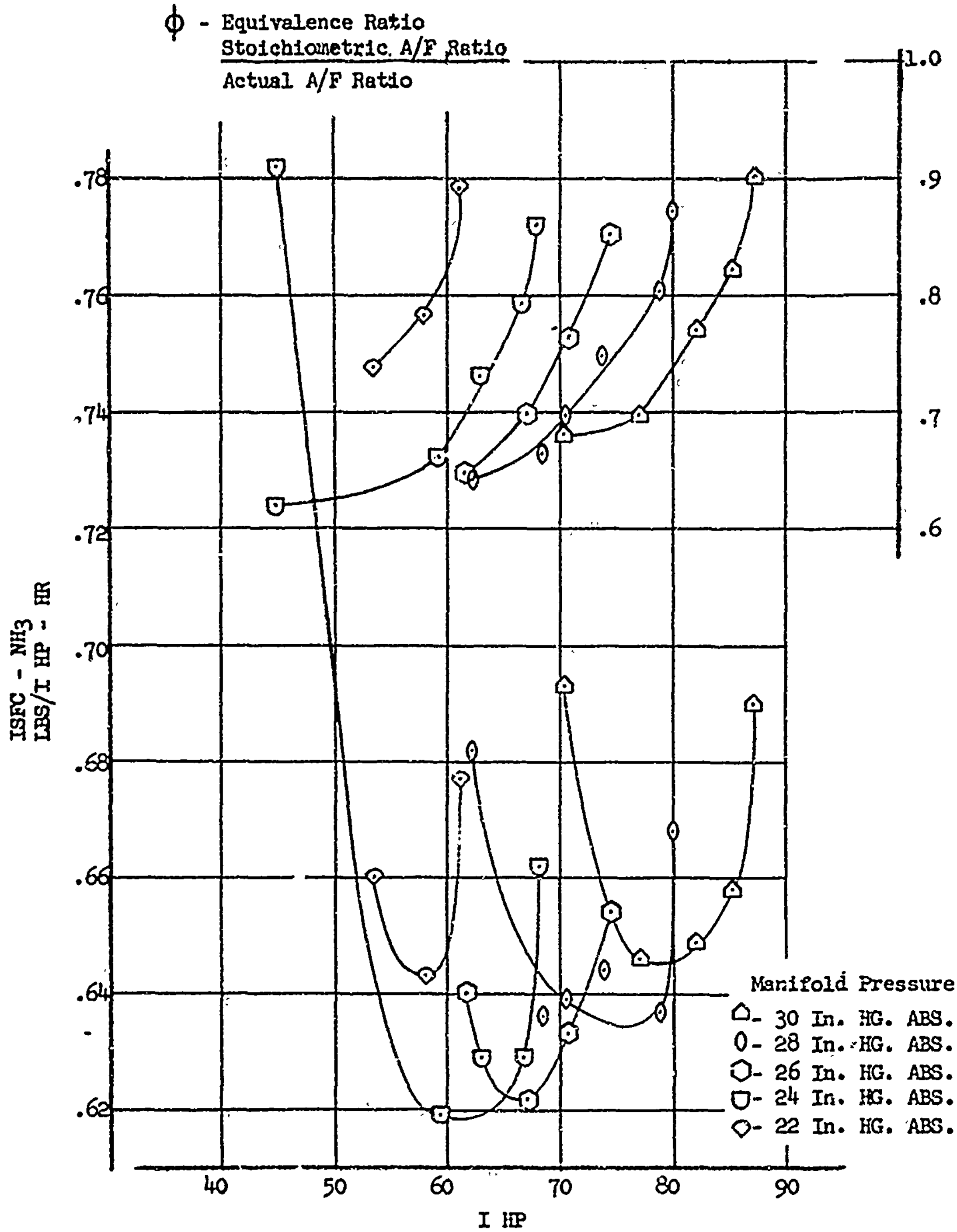



Fig. 71

NH3 - 232

AVDS-1790 V-Twin #3

ISFC and Equivalence Ratio vs. Indicated Horsepower on $NH_3$ Vapor with Mallory Mag., R-120 Plugs, Gap .070", 12:1 CR @ 1800 RPM

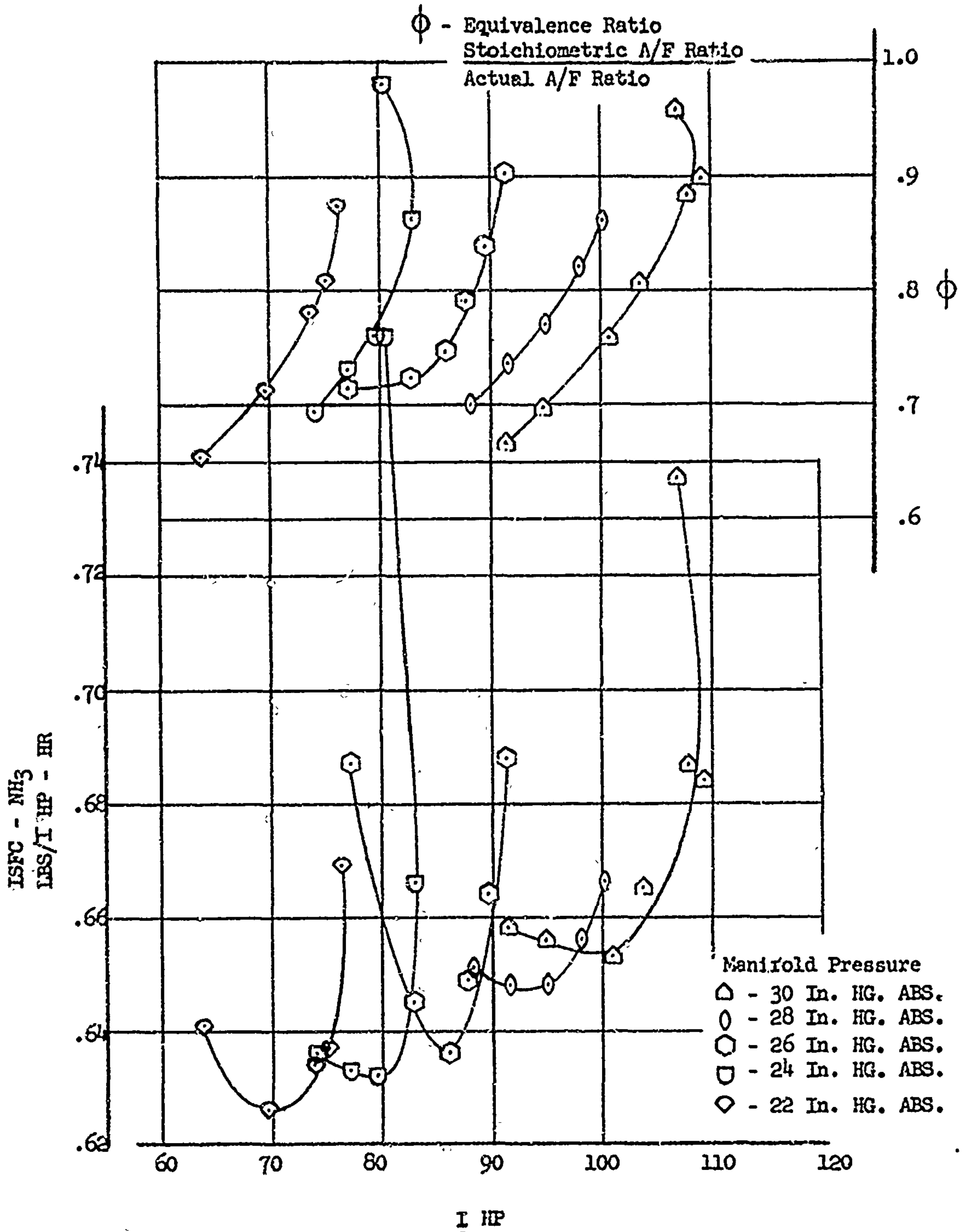





Fig. 72

AVDS-1790 V-Twin #3

ISFC and Equivalence Ratio vs. Indicated Horsepower on $NH_3$ Vapor with Mallory Mag., R-120 Plugs, Gap .070", 12:1 CR @ 2100 RPM

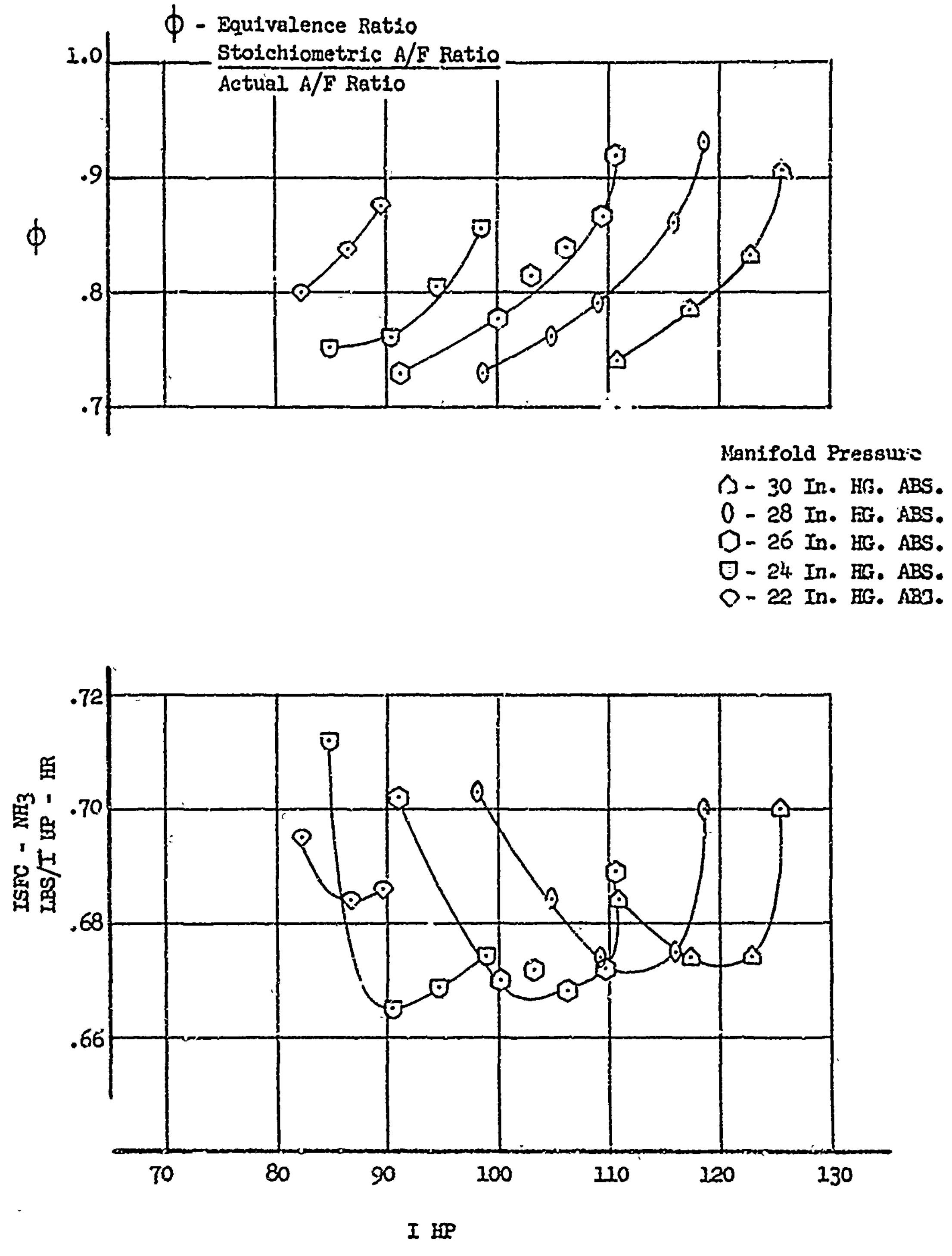





Fig. 73

AVDS-1790 V-Twin #3

ISFC and Equivalence Ratio vs. Indicated Horsepower on $NH_3$ Vapor with Mallory Mag. R-120 Plugs, Gap .070", 12:1 CR @ 2400 RPM

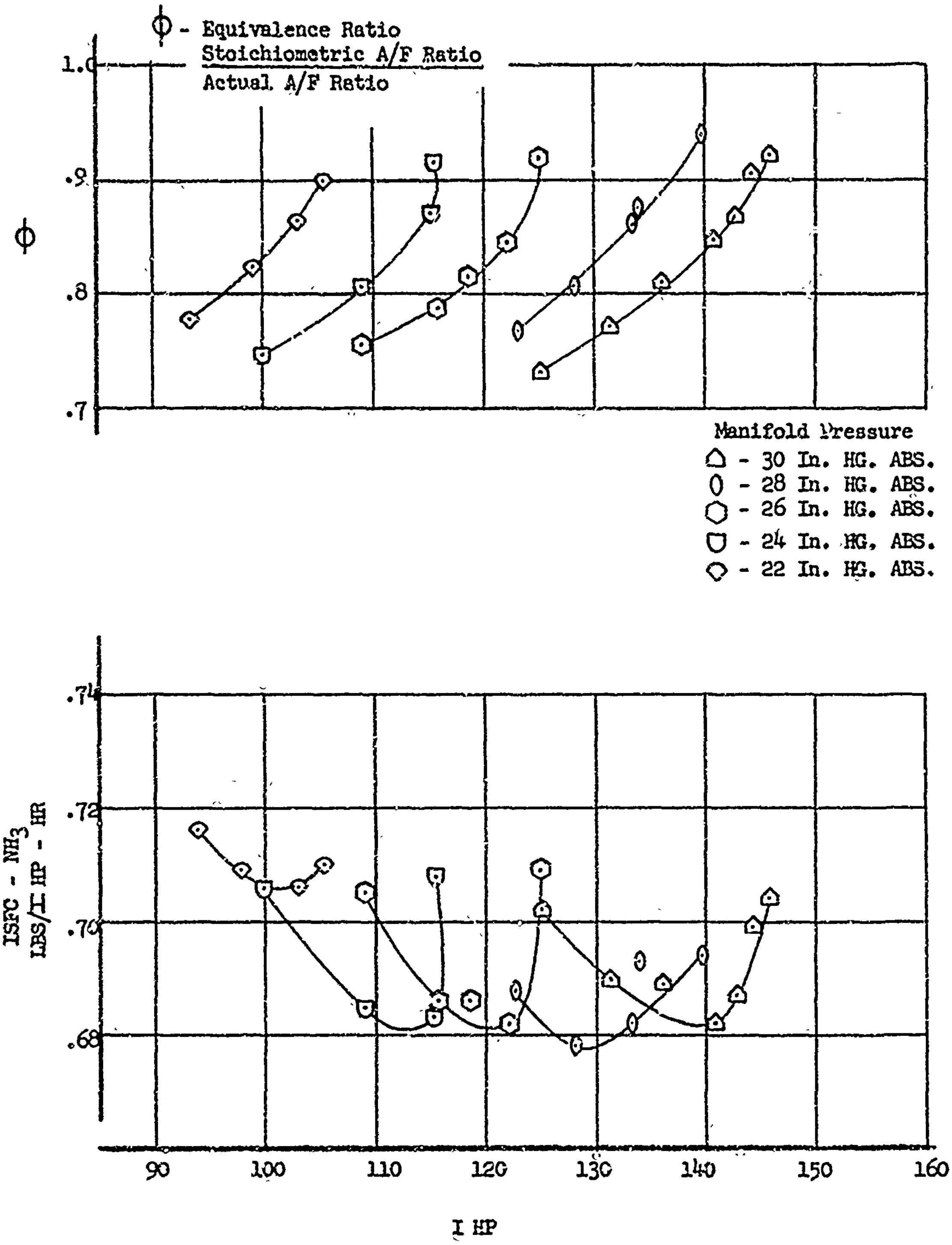





Fig. 74

NH3-237

AVDS 1790 V-Twin #3

Effects of compression ratio on ISFC and Peak Pressures at optimum ISFC points -vs- manifold pressures @ 1800 RPM.

H. VanSweden

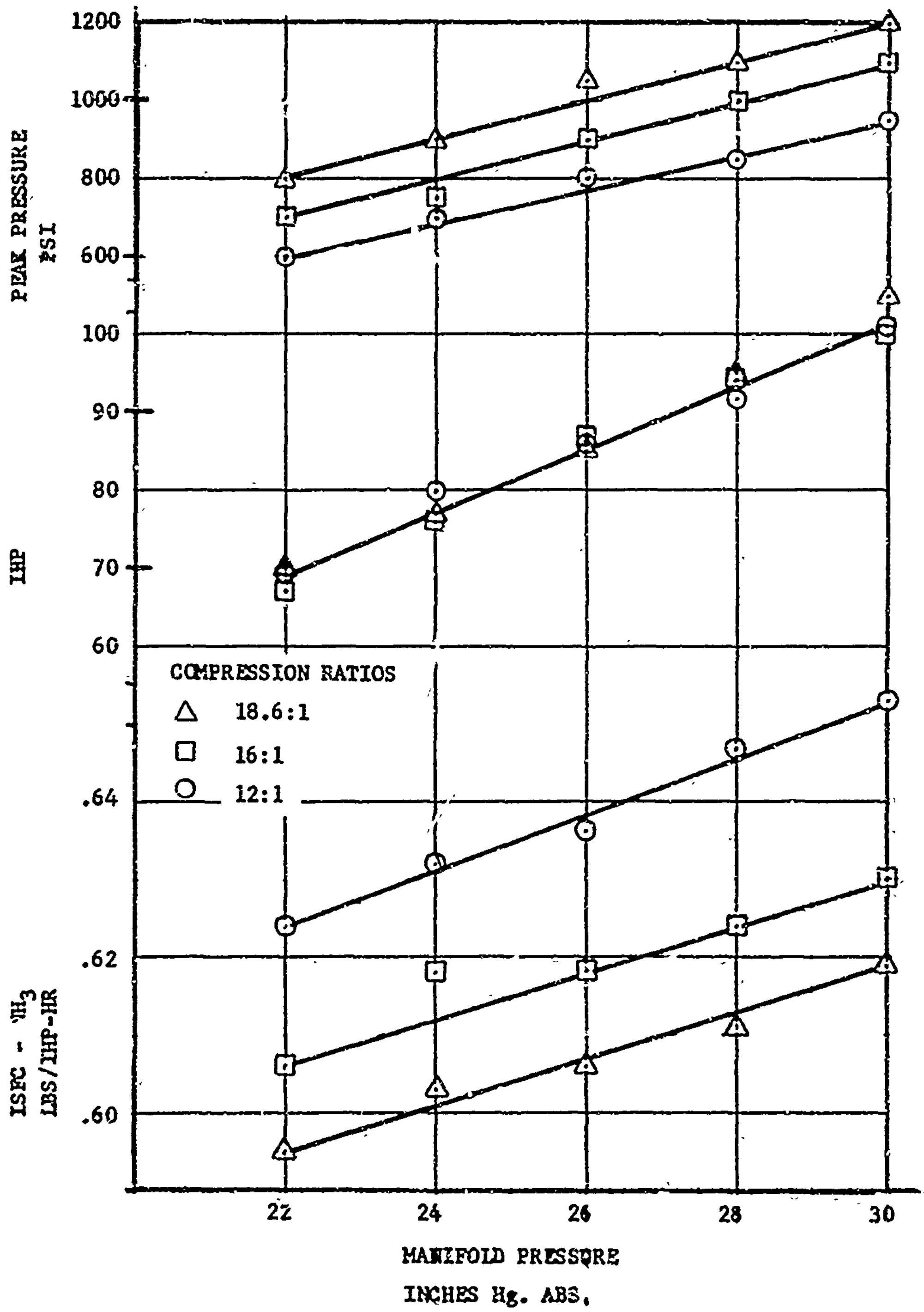





Fig. 75

# DISCUSSION

3. Horsepower output is greater with spark-ignition than with either pilot ignition or straight diesel operation. It is possible to obtain supercharged diesel output with unsupercharged ammonia fuel spark-ignition.

4. Neither compression ratio nor ignition source appears to have any significant change in the equivalence ratio at which either best fuel economy or best power is obtained.

5. Figures 32 and 76 show brake specific fuel consumption maps of the four cylinder, 3-7/8-inch bore L-141 engine and the two cylinder, 5-3/4-inch bore 1790 engine. It is readily apparent that the large bore is no handicap in operating with ammonia fuel; in fact, there is a considerable improvement in fuel consumption with the larger cylinder. This may well be due to the more concentrated combustion chamber and the more centrally located spark plug.

6. The spark-ignition engine has inherent advantages over the pilot fuel engine; namely its ability to run throttled at part-load, no need for a secondary fuel, and more reliability of the ignition source. Development of fuel injection systems for handling extremely small quantities of pilot fuel could eliminate the last item.

## ADDITIVES

A subcontract was let to the Research and Development Department of American Oil Company in Whiting, Indiana, to investigate the effect of chemical additives to promote ignition and combustion of anhydrous ammonia in both spark-ignition and compression-ignition engines. To provide information on the ammonia combustion process and thus on the mechanism by which an additive might function, a bench reactor was devised which would facilitate the measurement of the ignition temperature and level of oxidation of a mixture of ammonia and oxygen, or ammonia and oxygen plus an additive.

### Selection of Additives

Additives were selected according to possible use with ammonia in spark-ignition engines and in compression-ignition engines. For spark-ignition engines, the primary interest was in gaseous additives which could

AVDS 1790 V-Twin #3
Fuel Consumption Map - Performance on $NH_3$ Vapor with Mallory Magneto, R-115 Plugs, Gap .055" 18.6:1 CR

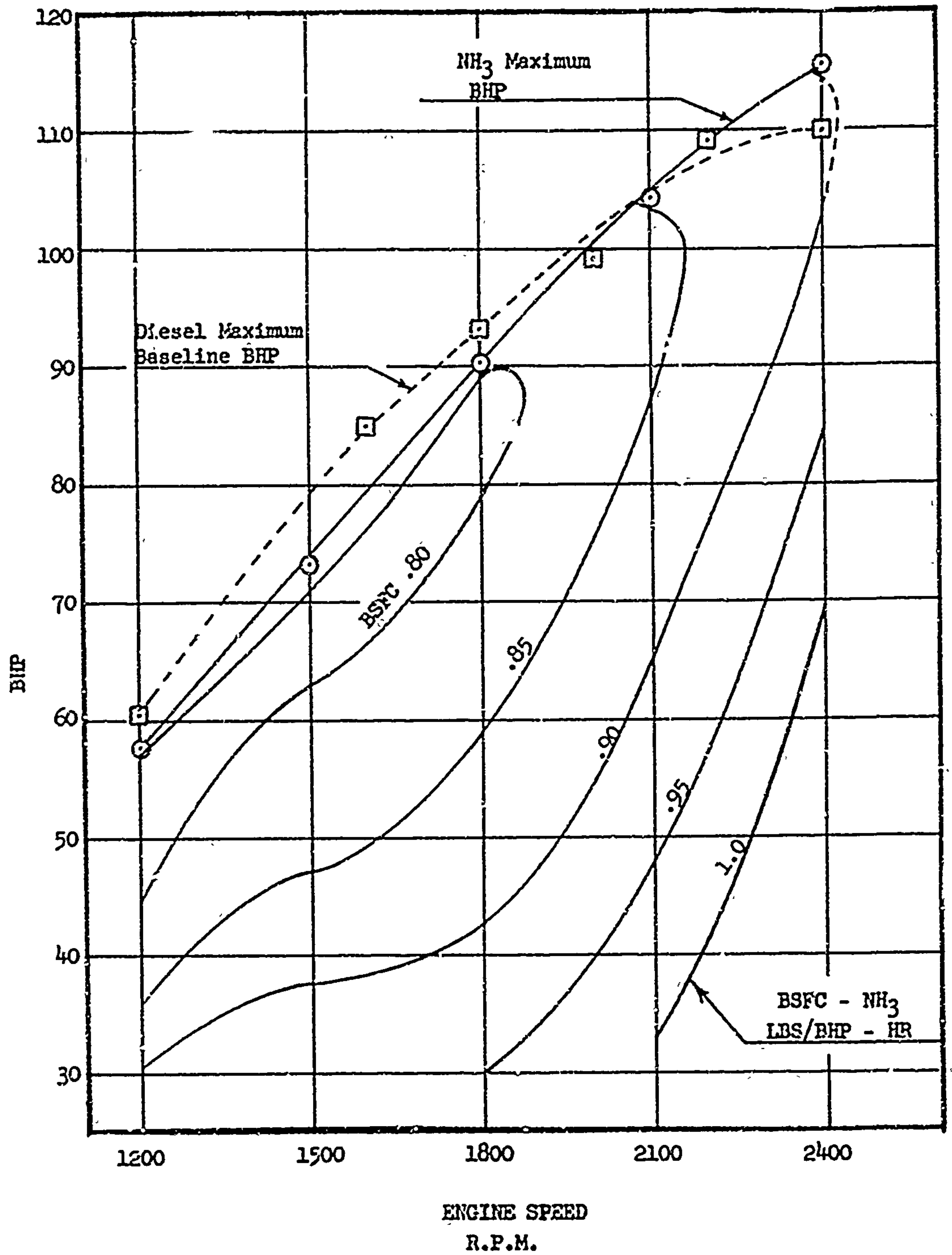





Fig. 76

## DISCUSSION

be introduced with gaseous ammonia, although some liquid additives were considered. For compression-ignition engines, the primary interest was in additives that might be effective at low concentrations and that would be soluble in liquid ammonia, although gaseous additives were also considered for introduction via the intake air.

### Bench Tests

Eight gases and two liquids were tested in the bench reactor as additives for spark-ignition engines. Two of the gases, hydrogen and ozone, appeared to have a strong effect on the oxidation of ammonia, whereas the remainder had slight or no effect. Both liquids, hydrocarbons, had strong positive effects on oxidation of ammonia. The scope of work on additives for spark-ignition engines was limited because the favorable performance of hydrogen and acetylene was well known.

A total of 45 additives were tested in the bench reactor as additives for compression-- ignition engines. All were considered to be soluble in liquid ammonia, and several were tested at various concentrations. Of this total, on [illegible]typhnic acid, was found to induce ignition at concentrations as low as 0.[illegible] percent by weight of ammonia. Six additives at one-half percent substantially reduced ignition temperatures as did six additives at 1.0 percent, two at 1.5 percent and four at 5.0 percent. Although not inducing ignition, 11 additives at 5.0 percent, or less, significantly increased the oxidation of ammonia.

## ENGINE TESTS

Ten gases and 14 liquids were tested as ammonia additives in the spark-ignition CFR engine at 900 and 1800 rpm and 8.0:1 compression ratio, 35 degree spark advance and best power air-fuel ratio. Engine performance on ammonia only and on hydrogen only was checked first for reference purposes. Of the gases, hydrogen was by far the best additive, permitting fairly good engine operation at 900 and 1800 rpm at concentrations of less than 1.5 percent by volume. Acetylene was next best, requiring about six percent to give good engine operation at both speeds. The liquid additives, although several permitted good engine operation at both 900 and 1800 rpm, all required use of unreasonably high (over 10 percent) concentrations.

## DISCUSSION

Compression-ignition engine tests consisted of an evaluation of the performance of gases added to the intake air, and of several liquid-ammonia additives selected from the bench reactor program. The work on the inducted gases was conducted first, to simultaneously evaluate the gases as ammonia additives and pinpoint the best engine operating conditions to use for evaluating the liquid-ammonia additives.

Of gases added to the intake air while ammonia was being injected, hydrogen, normal butane, and acetylene all facilitated combustion at some engine operating condition. The gases generally were tested at 20:1, 25:1 and 30:1 compression ratio, and at 900 and 1800 rpm. Combustion with ammonia and hydrogen was achieved at 25.1 and at 30.1 compression ratio at both 900 and 1800 rpm, performance improved as either compression ratio or speed was increased. Combustion with ammonia and normal butane was achieved at all compression ratios and speeds (not tested at 30:1 cr and 1800 rpm), but butane consumption was excessive at 1800 rpm. Combustion with ammonia and acetylene was achieved at 20:1 and 25:1 cr and 900 and 1800 rpm (not tested at 30:1 cr). Performance improved as either compression ratio or speed was increased.

The additives selected from the bench reactor program were engine tested at 25.1 cr and 900 rpm, the condition where all inducted gases performed most similarly. Of the 'non-explosives", ammonium nitrate was tested at concentrations of 1.0 and 5.0 percent of an additive-liquid ammonia mixture, and ammonium perchlorate was tested at 1.0 percent. Styphnic acid was selected as the most effective "explosive" and was tested at 0.1 percent. During the testing of these additives, the engine was closely observed for any evidence of the additive affecting combustion, and was checked for fire-ability by bracketing runs on the additives with runs using inducted acetylene. All of these liquid-ammonia additives proved to be completely without discernible tendency to initiate combustion of ammonia in this engine.

A detailed report of the additive investigation, including curves, tables and photographs is included as Appendix I of this report.

### FLAME TUBE TESTS

It was originally intended to evaluate the most promising additives uncovered in the investigation by AMOCO by running flame stabilization tests in the Allison flame tube. Prior to the completion of AMOCO's investigation, Continental authorized Allison to conduct screening tests on three oxides of nitrogen, carbon monoxide, ammonium nitrate and hydrogen peroxide. All of these materials except carbon monoxide were potentially producible in the field by the Energy Depot concept. Since the results from AMOCO did not produce any practical additives

DISCUSSION

other than hydrogen (which can be produced by partially dissociating ammonia) this list was never augmented.

For this screening effort, all gaseous additives were checked at a concentration of five percent by volume of fuel flow. Stable burning was established and the fuel and additive flows adjusted to the desired test condition. The air-flow rate was then increased or decreased until blow-out resulted.

The results of these tests are summarized as follows:

1. In five percent volume, none of the additives tested increased the apparent ammonia-air flame propagation velocity to the degree obtainable by dissociating 28 percent of the incoming ammonia fuel (five percent hydrogen by weight).

2. The beneficial effect of carbon monoxide and the nitrogen oxides are very dependent upon the local fuel-air ratio of the ammonia - air mixture.

3. Nitric oxide, NO, appears to offer the most promise as a flame propagation rate improver over the full equivalence ratio range of 0.9 to 1.2. However, it did not significantly exceed the effect of five percent acetylene additions at any mixture ratio. Also the formation of solid deposits resulting when the additive vapor in air mixed with the ammonia fuel could render its engine application difficult.

4. Carbon monoxide, CO, provided significant improvement to the ammonia - air flame propagation rate at lean air-fuel ratio (0.9 stoichiometric), but was ineffective at fuel-rich conditions.

5. Flame stability limits were extended considerably at fuel-rich ($>$1.1 of stoichiometric) conditions by nitrogen dioxide, $NO_2$ addition, but the benefits were insignificant in the stoichiometric range. Undesireable solid deposits were also experienced with this material.

6. No conclusive results could be obtained as to the effect of ammonium nitrate additions due to injection and mixing problems inherent to the test burner system.

## DISCUSSION

A detailed report of the investigation of the stable burning limits of ammonia - air flames, including sketches and graphs, is included as Appendix II of this report.

## IONIZATION

It had been shown in previous work that the combustion of ammonia-air mixtures could be initiated by spark devices. Ordinary spark devices, however, operating in pure ammonia and air mixtures left much to be desired, since the energy they made available was marginal for ignition. The purpose of this study was to learn whether or not high-energy line discharge by the longest possible path could alleviate these problems.

It was first assumed that if many times the minimum ignition energy were available, ammonia decomposition would (on a sufficiently large scale) result in the realization of better burning properties. Secondly, it was assumed that if much of the volume of the chamber were to be affected by ionization due to the discharge, rather than by the point source presented by a spark plug, that the low flame speed of ammonia would not necessarily be a critical consideration.

The investigation was conducted by means of a laboratory pulse generator which discharges its high energy by means of electrodes mounted in a "bomb" chamber containing controlled mixtures of ammonia and air at various pressures and temperatures.

1. Test Program. In order to test the pulse generator, atmospheric discharges with varying arc lengths were tried. Spectacular arcs up to 10 inches in length were drawn, Fig. 77. When of any length in excess of about four inches, the arc is not confined to a single main bolt, but rather is spread to many fine tendrils following the lines of the electric field.

   A series of "open-air" observations was made with the electrodes mounted in the head plates to determine the path of the arc. A strong tendency to "short-cut" through the head plate was noted.

   An effort was made to use hydrocarbon fuel, in the form of gasoline, for reference and calibration. Difficulty in controlling the air-fuel ratio and the degree of vaporization rendered this approach impractical, and hydrogen was used for check-out and calibration.

Fig. 77. High-Energy Pulses Discharging in Air.

## DISCUSSION

Ammonia combustion was then investigated. Tests were conducted to determine the effect of fuel-air ratio, initial pressure, discharge energy, temperature and spark gap on the combustion time and rate of pressure rise of ammonia fuel.

2. Results. In general, it was shown that the combustion time was affected by the parameters investigated as follows:

   a. Combustion time was optimum with an air/fuel ratio of 4.55 by volume or 7.63 by weight.

   b. Total pressure has very little affect on combustion time.

   c. Discharge energy has slight affect on combustion time.

   d. Combustion time decreases significantly with increasing temperature.

   e. There was a large difference in total combustion time, at all energy levels and temperatures, between small-gap and large-gap electrodes.

The rate of pressure rise is very strongly influenced by increases in spark gap, initial pressure and initial temperature.

The work on Ionization is reported in detail in Appendix IV.

## RADIO-FREQUENCY DISSOCIATION OF NITROGEN

The large ignition energy requirement and the low flame speed of ammonia-air mixtures suggested the idea that unconventional ignition means might be required for satisfactory combustion. One scheme that appeared to be attractive was that of dissociating a part of the incoming nitrogen in the intake manifold by means of an electrodeless Radio-Frequency (RF) discharge. The atomic nitrogen would then be distributed to the engine cylinders along with the ammonia-air mixture. Upon recombining, the nitrogen dissociation energy would be recovered and would be available for ignition of the ammonia. In this way, large amounts of ignition energy could be added to the ammonia-air mixture and, most importantly, the energy would be distributed over the entire volume of reactants, instead of being concentrated in a small region of space around the spark discharge.

## DISCUSSION

Before building any hardware or initiating any experimental tests, it was decided that theoretical calculations should be made to determine:

1. The time scale of the nitrogen recombination as a function of temperature and initial atomic nitrogen density.
2. Significant effect of the presence of oxygen on the time scale of the recombination.

These questions have been examined by solving appropriate sets of differential equations that express the rates at which the various chemical reactions proceed.

### Results

Recombination of atomic nitrogen to molecular nitrogen takes place at an extremely rapid rate; for example, under typical conditions (standard temperature and pressure, 10 percent dissociation) approximately 99 percent of the nitrogen has recombined in less than 0.1 millisecond. It would appear, because the major part of the recombination will occur before the dissociated nitrogen reaches the engine cylinders, nitrogen dissociation is not a reliable ignition source.

A detailed report of the RF dissociation investigation, including curves, tables and photographs is included as Appendix IV of this report.

## RADIO-FREQUENCY DISSOCIATION OF AMMONIA

In developing a means of operating engines using anhydrous ammonia as a fuel it was determined that performance of the spark-ignition engine was enhanced considerably (particularly at the high-speed end) by the addition of free hydrogen to the fuel-air mixture. One method of obtaining free hydrogen is by the thermal dissociation of ammonia. Experience with thermal dissociation, as reported above, has shown that operation is satisfactory when sufficient hydrogen is generated. However, during starting, low load and certain transient conditions, the production of hydrogen is insufficient. It was therefore determined that an investigation of other means of dissociating ammonia was in order.

## DISCUSSION

Analytical studies were initiated at Stevens Institute of Technology to establish the feasibility of ammonia dissociation by use of radio frequency (RF) energy. A literature survey relative to RF dissociation of ammonia was conducted to determine the reaction kinetics and rate coefficients. Mathematical models of the reaction system were established and sets of equations were solved by use of a high-speed digital computer. Since the reactions are temperature dependent, solutions were obtained at $0^o$, $20^o$, $50^o$ and $100^o$ centigrade.

The results of this study indicate that RF dissociation of ammonia is impractical since the energy required to generate one-half of one percent hydrogen would require approximately 50 percent of the engine output. A detailed report of the investigation of RF dissociation of ammonia, including curves and tables, is included as Appendix V of this report.

## MATERIAL COMPATABILITY

In only one case in the performance of this contract was any part adversely affected by the use of anhydrous ammonia as a fuel; copper spark-plug gaskets, installed in the engine as delivered, completely dissolved after a very short period of operation. Soft iron gaskets were used as replacements and no further trouble was encountered.

Figure 78 shows typical deposits found on the piston pin crowns. Analysis of the deposits showed they consisted primarily of carbonaceous organic matter and oxides of barium, phosphorus and sulfur leading to the conclusion that they were formed from oxidation of the lubricating oil. The piston skirts, Fig. 79, were clean and in excellent condition.

Figures 80 and 81 show the condition of the connecting rod and main bearings after 297 hours of operation with ammonia fuel. It is interesting to note that the ammonia did not attack the copper in the copper-lead bearings nor in the aluminum piston alloy.

In general, it can be said that only pure copper or brass or bronze fittings are vulnerable to attacks by ammonia, and these items should be replaced by iron or other suitable materials.

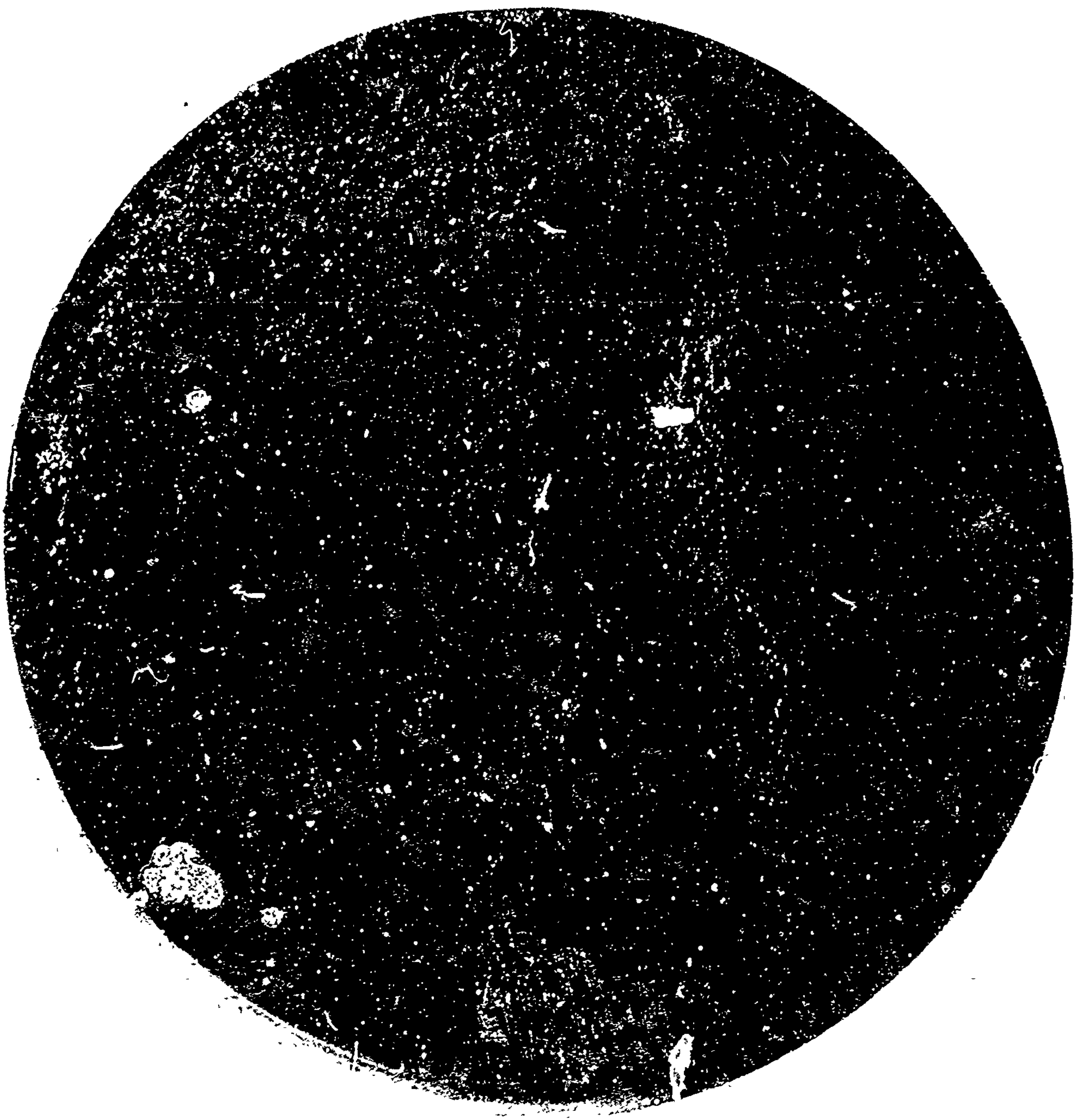


Fig. 78. Piston Deposits After 351 Hours of Operation, 297 Hours On Ammonia. (D-34120)

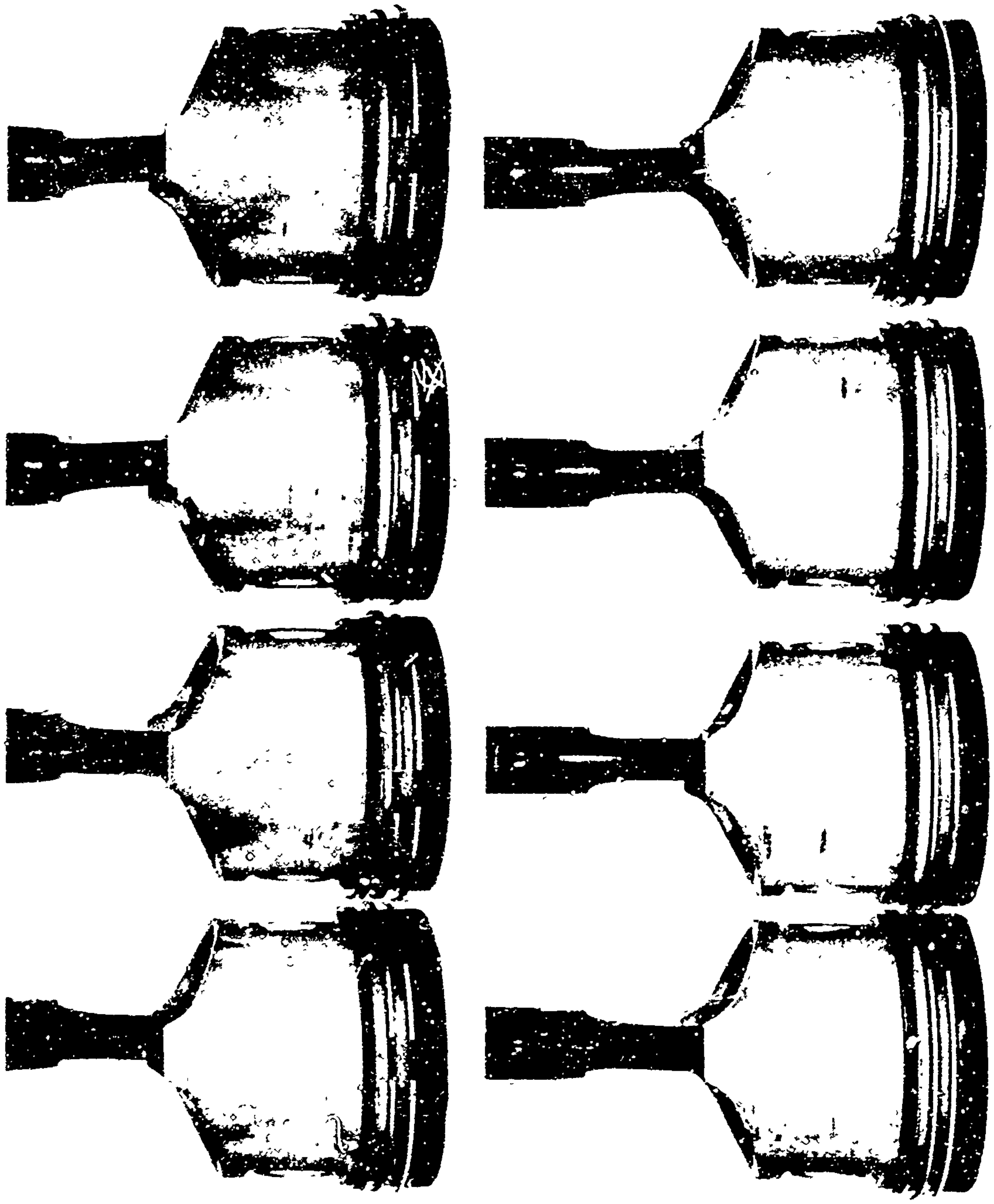


Fig. 79. Piston Skirts After 351 Hours of Operation. 297 Hours On Ammonia.
(D-34123)

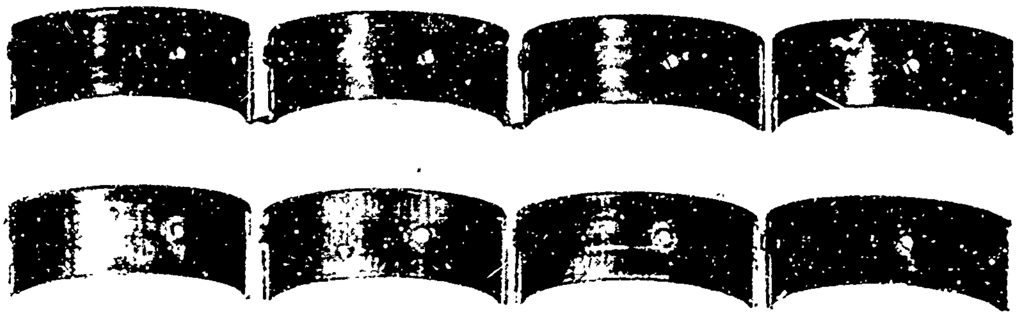


Fig. 80. Connecting Rod Bearings After 351 Hours Of Operation, 297 Hours On Ammonia. (D-34121)

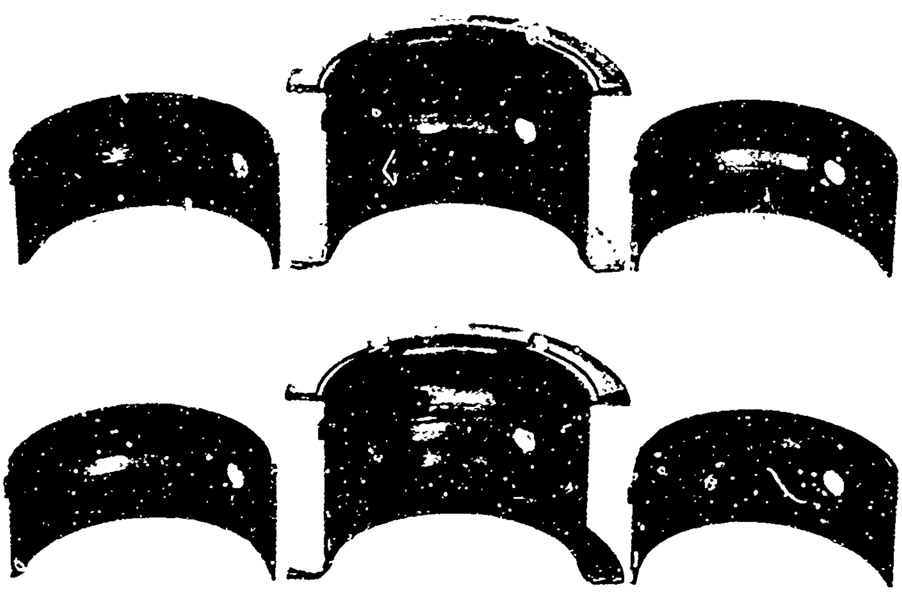


Fig. 81. Main bearings after 351 Hours Of Operation, 297 Hours An Ammonia. (D-34122)

DISCUSSION

## CONCLUDING REMARKS

The results of this investigation have shown that it is indeed feasible and practical to convert and operate both spark-ignition and compression-ignition engines with anhydrous ammonia fuel. Whether or not such a fuel can be justified from either an economic or military point of view is another question. From the price aspect, ammonia costs approximately ten times as much as hydrocarbon fuels for the same amount of energy, when both are bought in bulk quantities and tax free. It therefore becomes apparent that ammonia would never be used if adequate stocks of hydrocarbon fuels were readily available.

There are however two situations, resulting from military operations, where very strong cases can be made for use of anhydrous ammonia as a fuel. First is the strictly military situation in remote areas where supplying daily requirements of hydrocarbon fuels is an extremely difficult and highly expensive operation. Under these conditions, use of a fully developed Mobile Energy Depot for local generation of ammonia fuel would appear to be a logical and practical solution.

The second situation also results from military operation; when a local condition exists where the military, as a matter of expediency, has appropriated all existing stocks of hydrocarbon fuels, leaving nothing for use by the civilian economy, as happened in France and Belgium under German occupation during World War II. If this situation should be repeated in the future in a friendly country, or in the United States itself, due to loss of overseas sources of petroleum products, the ability to operate civilian vehicles on ammonia could make a tremendous contribution to the war effort.

Commercial sources of ammonia that could be diverted for this purpose have been expanding rapidly in recent years. Chemical Economics Handbook, Reference 8, shows a total synthetic ammonia capacity for the continental United States for the following years:

| January 1, | Thousands of short Tons $NH_3$ |
|---|---|
| 1964 | 7,838.3 |
| 1965 | 8,595.0 |
| 1966 | 10,540.8 |
| 1967 | 15,092.8 |
| 1968 | 17,246.1 |

# DISCUSSION

This report has shown methods of converting existing compression-ignition and spark-ignition engines to operate successfully using anhydrous ammonia as a fuel. It is considered that even better performance could be obtained from an engine that was specifically designed for operation on ammonia. Such an engine should embody at least the following design features:

1. It should be a spark-ignition engine utilizing a high energy ignition source, such as a magneto.

2. It should have a relatively high compression ratio of the order of 12:1 to 16:1.

3. The combustion chamber should be highly compacted, approaching a spherical shape.

4. Spark plug design should be of an extra long reach type so as to locate the spark gap near the center of mass of the combustion chamber.

5. A mild degree of supercharging should be provided to ensure a good horsepower to weight ratio.

6. Components should be designed to operate with peak cylinder pressures of at least 1200 psi.

7. If engine size is such that the rated speed will exceed 3000 rpm, a dissociator to provide up to 1.5 percent hydrogen should be used.

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